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TABLES

1.0 INTRODUCTION

Fluid film bearings have recently emerged as a viable technology for cryogenic turbopump applications. These bearings offer the promise of higher speeds, lower weights, and more robust designs. In order to advance the technology of these bearings so that their full potential can be realized, extensive research is required in the areas of flow visualization, flowfield measurement, and dynamic characteristics. The successful completion of such a research program depends on developing the right research tool (tester) and having the right facility to use it in. This study investigated the capability of three existing testers in order to determine the suitability of each for performing the required tasks. The testers considered are discussed below.

1.1 HYBRID BEARING TESTER

Figure 1-1 is a sketch of the Hybrid Bearing Tester (HBT). This tester was developed under a NASA Lewis contract (Winn et. al., 1974) for the testing of hybrid ball/hydrostatic bearings in liquid hydrogen. The tester has a hydrostatic loader, hydrostatic thrust bearing, and turbine drive. During testing (Spica and Hannum, 1986), it achieved a speed of 80,000 in liquid hydrogen and completed 337 start transient tests. Pressure and temperature measurements were made at various locations in the tester and in the bearing. Deflections were measured using fiber optic sensors. Accelerometers were mounted on the housing. No measurements were made of dynamic forces.

1.2 OTV BEARING TESTER

The OTV Bearing Tester is shown in Figure 1-2. This tester was designed under NASA Lewis contract (Anon, 1992) for high speed operation, unbalance testing, and transient start testing. The tester utilizes an integral Terry turbine drive, dual hydrostatic bearing support, and axial hydrostatic thrust bearings. It was designed for liquid hydrogen use. Pressure and temperature measurements can be made at various locations in the tester, but not in the hydrostatic bearing. Shaft motion can be measured using both fiber optic and eddy current proximity probes. Housing accelerations are measured using accelerometers. No dynamic force measurements can be made directly.

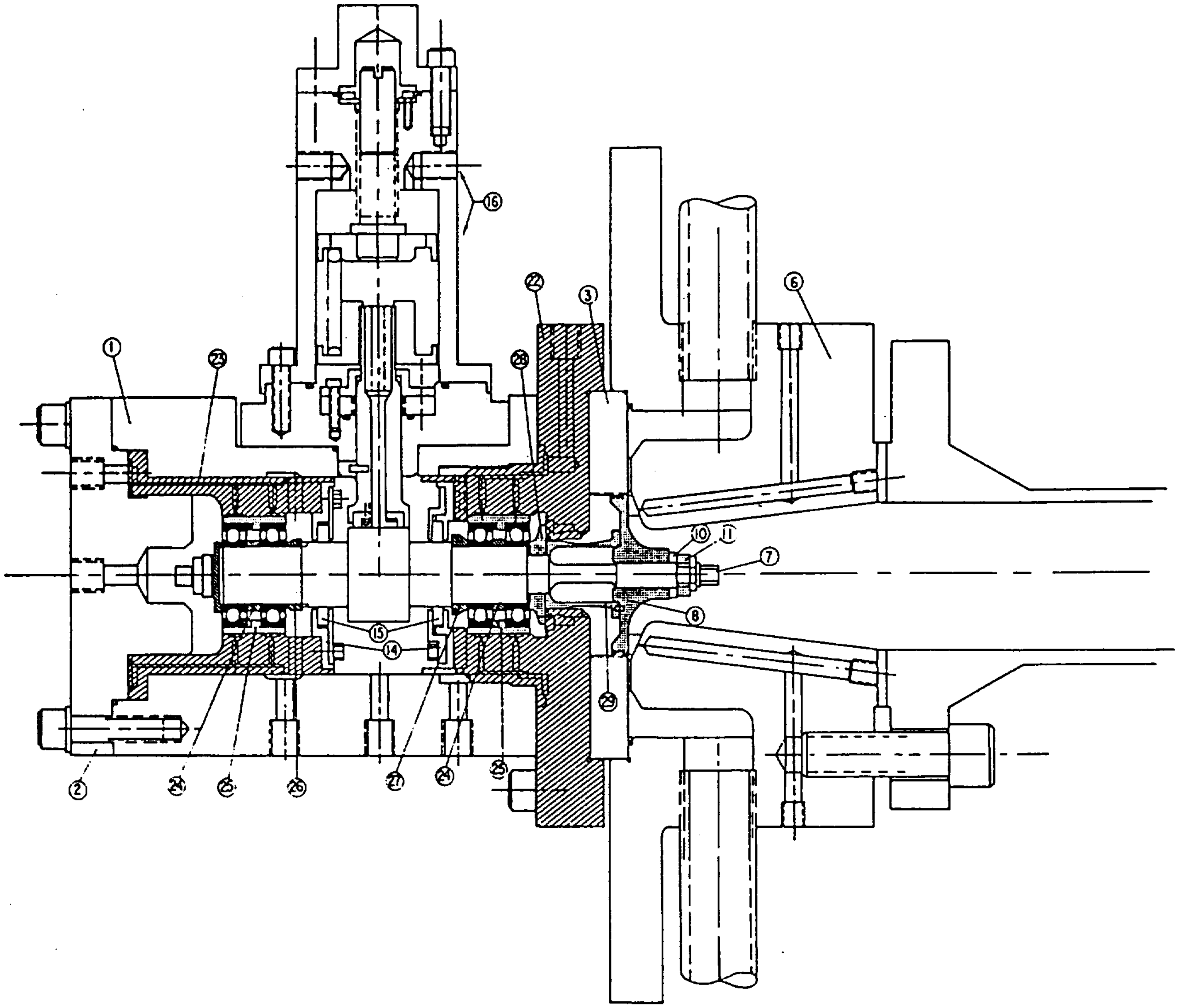


Figure 1-1. Hybrid Bearing Tester

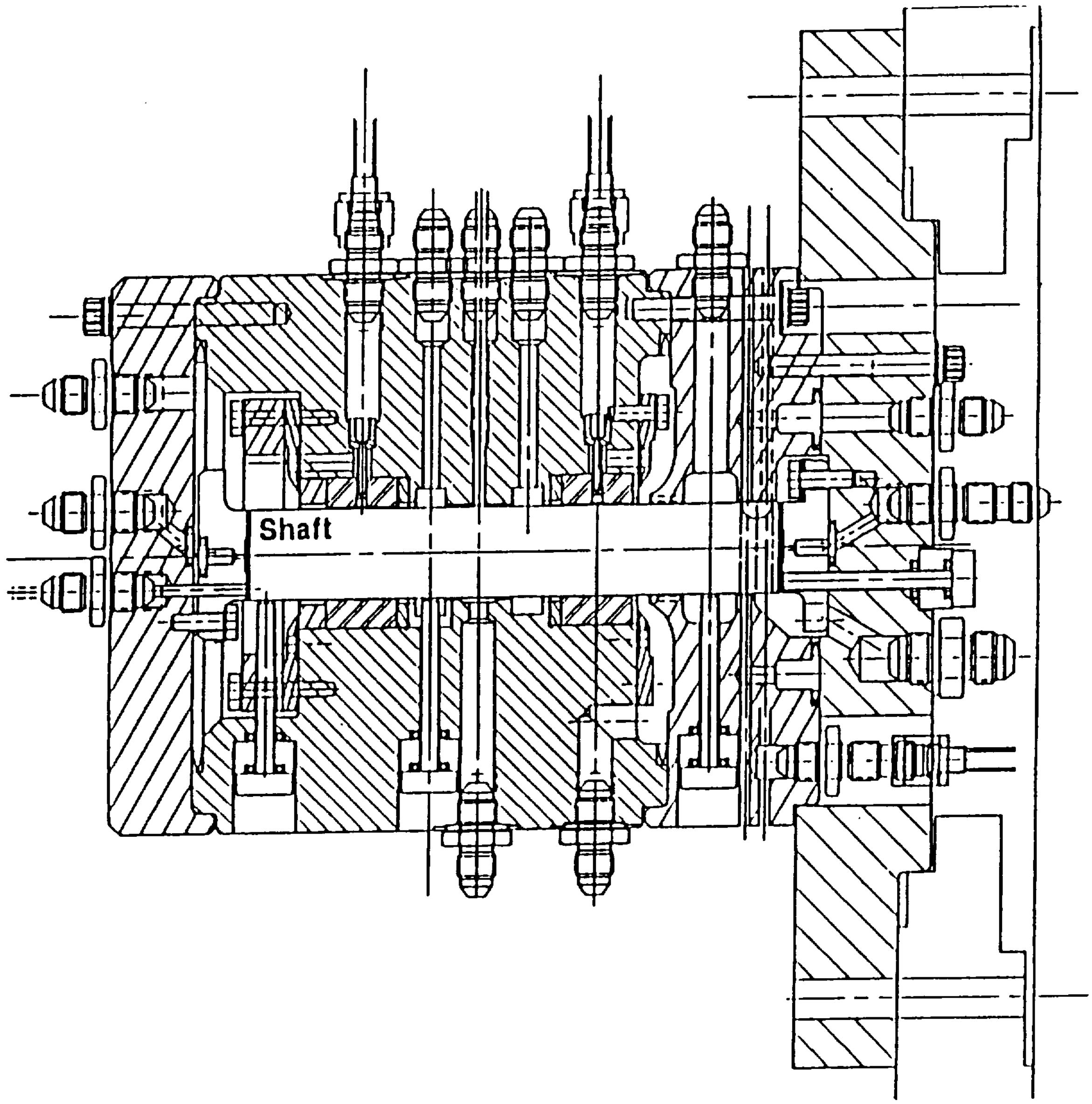


Figure 1-2. OTV Bearing Tester

1.3 LONG LIFE BEARING TESTER

The Long Life Bearing Tester (LLBT) is shown in Figure 1-3. This tester was originally developed under a NASA Lewis contract (Butner and Murphy, 1986) to study hybrid ball/hydrostatic bearings. Testing has been completed in liquid nitrogen (Scharrer, et al., 1991a&b), liquid hydrogen, freon, and liquid oxygen (Scharrer, et al., 1992a). It has been used more recently to test annular seals and orifice compensated hydrostatic bearings. This tester has been used in many configurations to perform thrust bearing transient testing, journal bearing/annular seal transient lift-off testing, and a failed attempt at extracting rotordynamic coefficients. Pressure and temperature measurements can be made at various locations in the tester and in the bearing bore. Shaft motion can be measured using eddy current proximity probes. Acceleration and dynamic force measurements can be made on the bearing carrier.

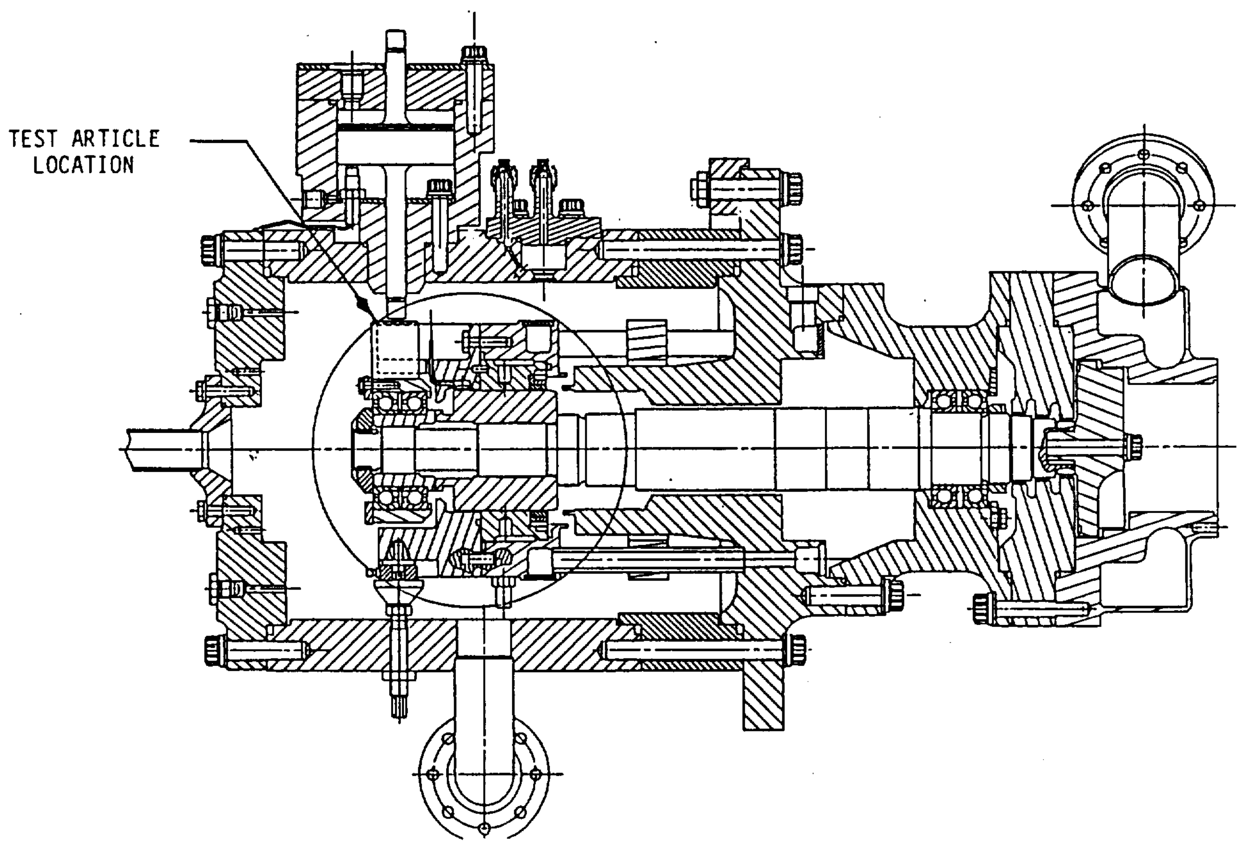


Figure 1-3. Long Life Bearing Tester

2.0 CRITERIA FOR EVALUATION

In order to provide some guidance for the evaluation of the candidate tester, a criteria was developed which addressed all major issues related to fluid film bearing testing: test article configurations, test fluid, instrumentation, and test objectives.

2.1 TEST ARTICLE CONFIGURATIONS

For the purposes of this evaluation and future test plans, only fluid film and, to a lesser extent, magnetic bearings will be considered. No solid lubricant or rolling element bearings are considered. The following bearing configurations were addressed specifically.

2.1.1 Hydrostatic Bearings

Hydrostatic bearings have shown great potential in recent NASA studies. Hydrostatic bearings come in two basic configurations: externally fed and internally fed. Both bearings require a substantial pressure drop across the orifice and land to provide the necessary rotordynamic characteristics and load capacity. A schematic of an orifice compensated, multi-recessed, externally fed, hydrostatic bearing is shown in Figure 2-1. Analytical tools for externally fed bearings are well developed (San Andres, 1990) and the database of experimental data is expanding rapidly (Kurtin et al., 1993). A schematic of an orifice compensated, multi-recessed, internally fed, hydrostatic bearing is shown in Figure 2-2. This bearing has limited application unless the turbopump is designed to accommodate it. This bearing holds great potential because of the simplified design required to provide high pressure to the bearings. Measurements required to validate analytical models include the following:

- Supply pressure and temperature
- Sump pressure and temperature
- Recess pressure and temperature
- Land pressure and temperature
- Shaft motion, eccentricity, and tilt
- Flow rate and shaft speed
- Load capacity
- Torque
- Rotordynamic force and moment coefficients
- Transient lift-off characteristics

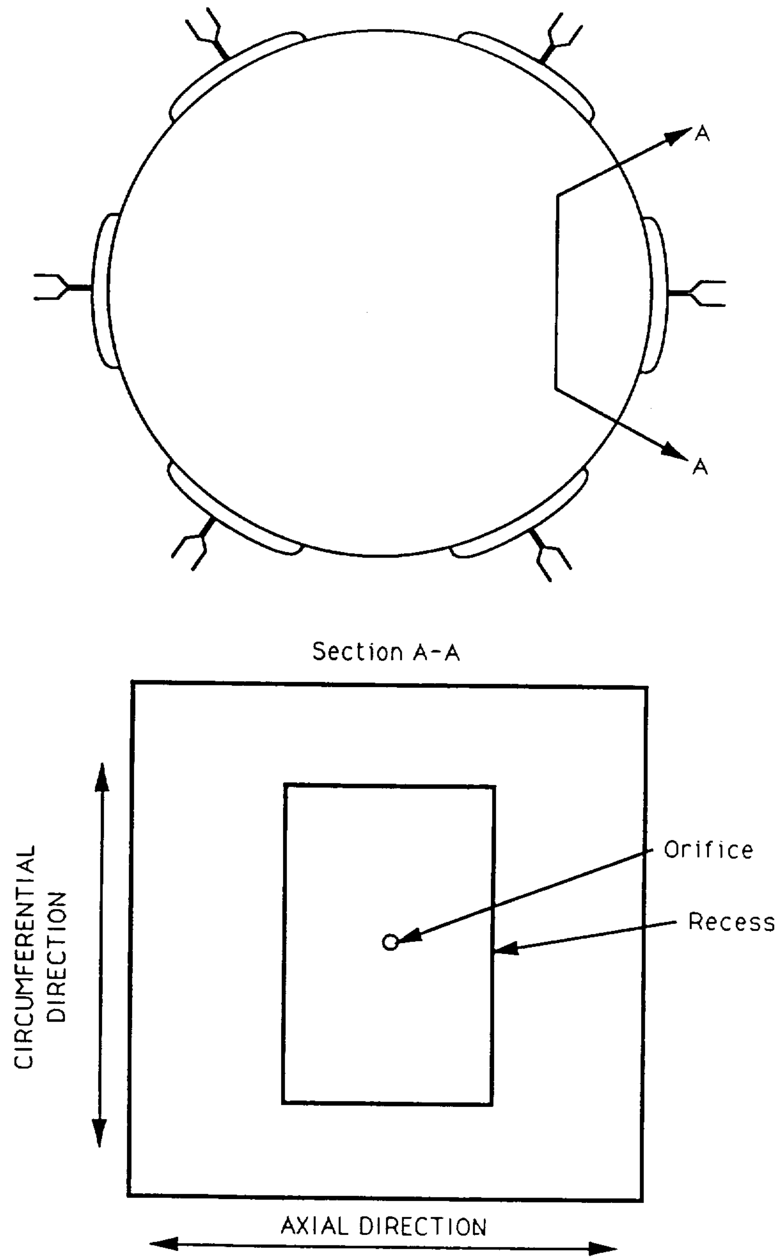


Figure 2-1. Externally fed hydrostatic bearing

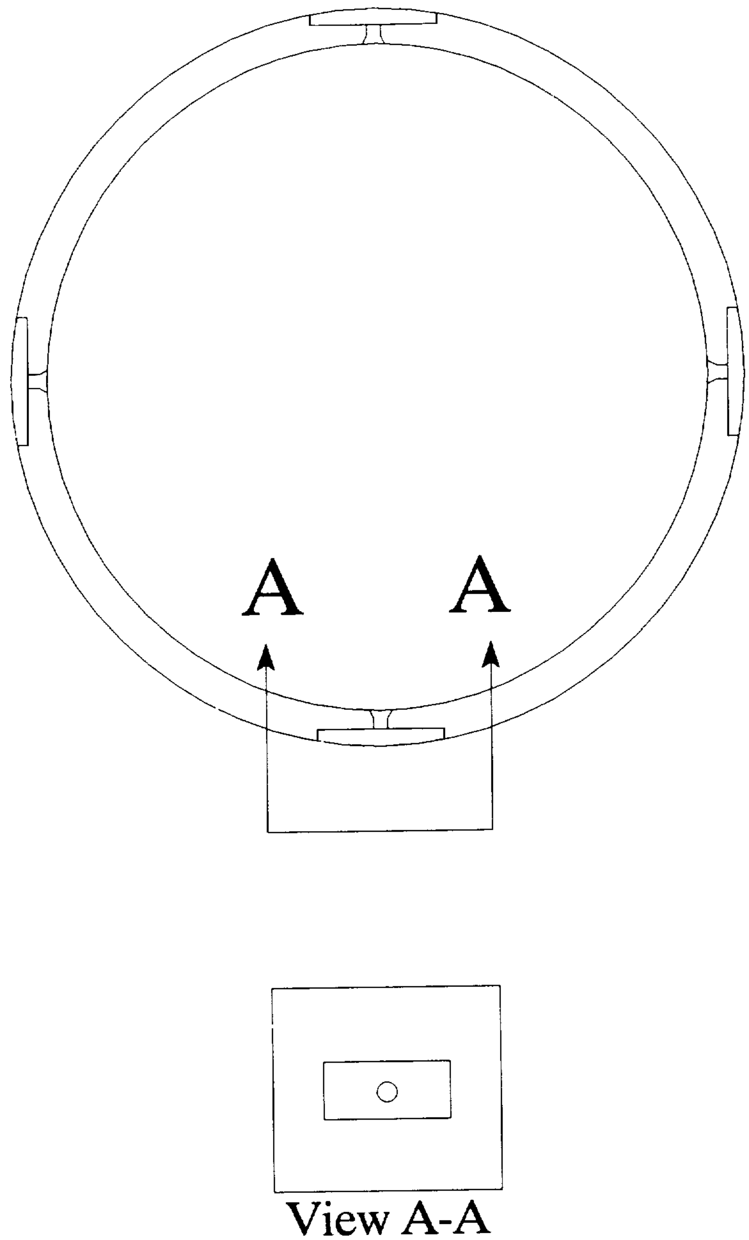


Figure 2-2. Internally fed hydrostatic bearing

2.1.2 Annular Seals (Damping Bearings)

Annular seals have been used in the recent past by NASA to serve as the primary rotor support bearing in the SSME HPOTP (Scharrer, et al., 1992b and Nolan et al., 1993). In the damper seal configuration these seals have attractive rotordynamic coefficients, load capacity, and leakage. Analytical tools for annular seals are well developed (San Andres, 1991) and a database of experimental data exists (Childs and Kim, 1985). A schematic of an annular seal is shown in Figure 2-3. Like the hydrostatic bearing, this element requires a substantial pressure drop in order to provide needed load capacity and rotordynamic characteristics. Since inlet boundary conditions dominate the characteristics of this device, provisions should be made for swirl brakes (radial vanes at the inlet) in any test program. Measurements required to validate analytical models include the following:

- Supply pressure and temperature
- Sump pressure and temperature
- Internal pressure and temperature profile
- Inlet fluid velocity components
- Shaft motion, eccentricity, and tilt
- Flow rate and shaft speed
- Load capacity
- Torque
- Rotordynamic force and moment coefficients
- Transient lift-off characteristics

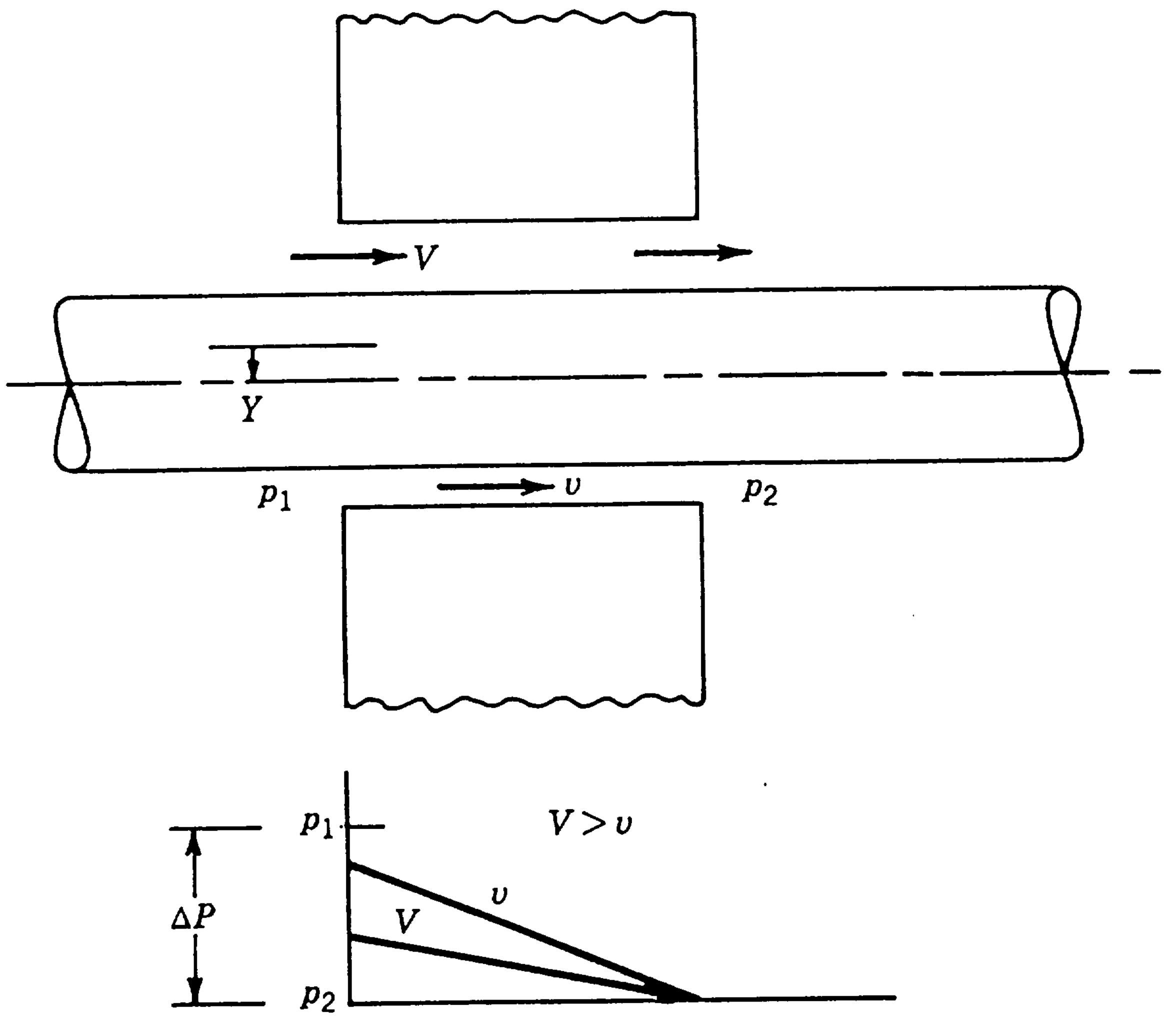
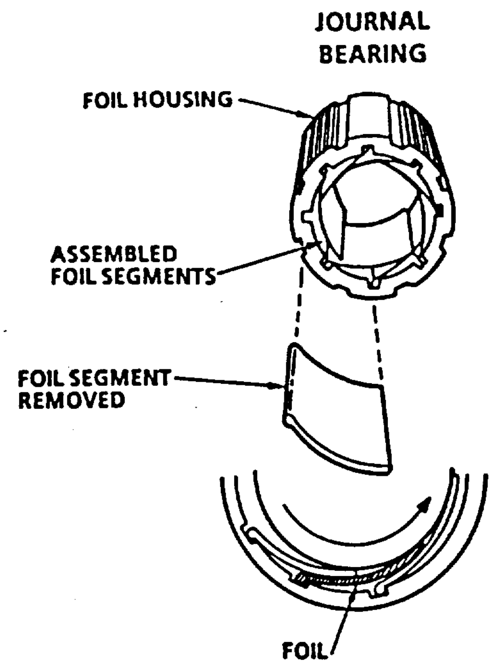


Figure 2-3. Annular seal

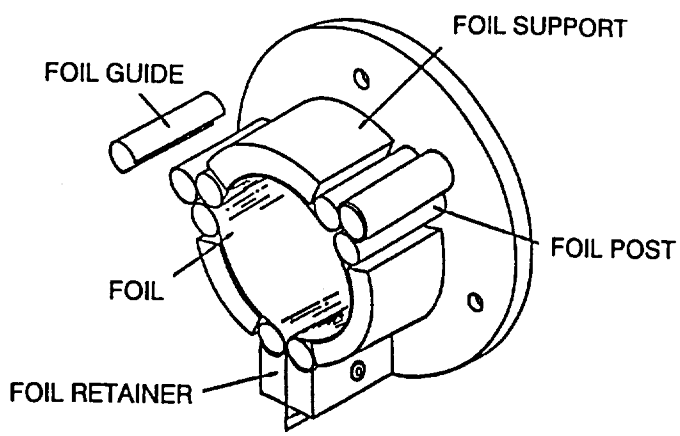
2.1.3 Foil Bearings

Foil bearings are a technology which hold great potential for turbopump applications. Recent test programs at NASA (Saville, et al., 1991) have demonstrated their capabilities in cryogens. Analytical tools are not well developed and there is a general dearth of experimental information. There are basically two different types of foil bearings: bending and tension. A schematic of the bending type of foil bearing is shown in Figure 2-4. This bearing requires a pressurized environment to insure high fluid density for maximum load capacity. The top and bump foils are coated with materials which provide high coulomb damping when rubbing occurs between them. A schematic of the tension type foil bearing is shown in Figure 2-5. A similar pressure environment is also beneficial to this type of foil bearing. The two types of foil bearings (bending and tension) have different tester requirements. The bending type of foil bearing typically has a large L/D ratio while the tension type of foil bearing typically requires more radial space. Measurements required to validate analytical models include the following:

- Supply pressure and temperature
- Sump pressure and temperature
- Internal pressure and temperature profile
- Foil deflected shape
- Shaft motion, eccentricity, and tilt
- Flow rate and shaft speed
- Load capacity
- Torque
- Rotordynamic force and moment coefficients
- Transient lift-off characteristics



bending type foil bearing



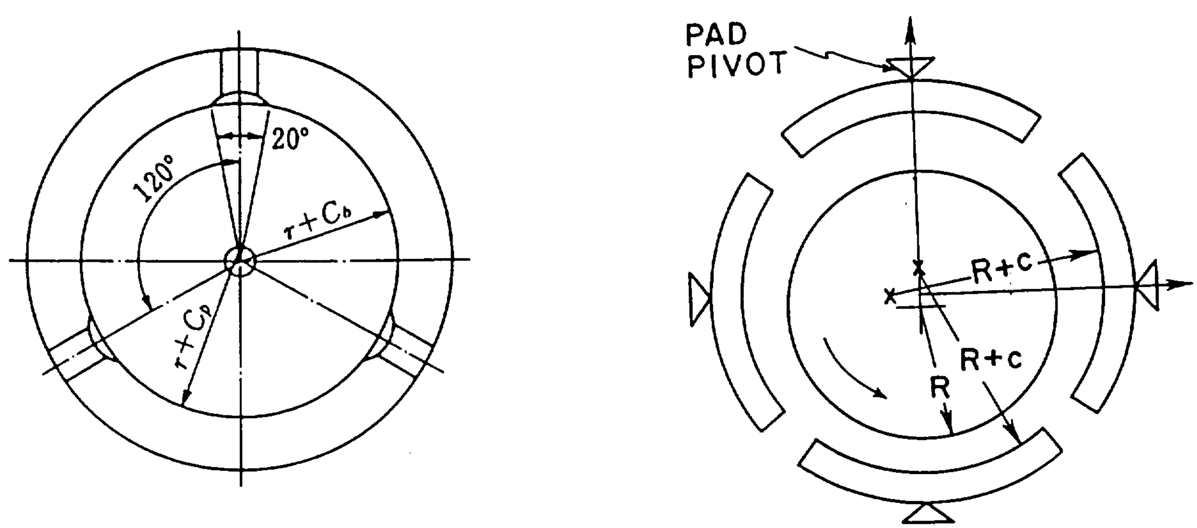
tension type foil bearing

Figure 2-4. Bending and tension type foil bearings

2.1.4 Hydrodynamic Fluid Film Bearings

Hydrodynamic fluid film bearing include fixed geometry bearings such as sleeve, axially grooved, multilobe, and pressure dam as well as variable geometry bearings such as tilt pad bearings. Sample hydrodynamic bearing geometries of both types are shown in Figure 2-5. This figure shows multilobe and tilt pad bearings. Measurements required to validate analytical models for hydrodynamic bearings are as follows:

- Supply pressure and temperature
- Sump pressure and temperature
- Internal pressure and temperature profile
- Shaft motion, eccentricity, and tilt
- Flow rate and shaft speed
- Load capacity
- Torque
- Rotordynamic force and moment coefficients
- Transient lift-off characteristics



multilobe bearing

tilt pad bearing

Figure 2-5. Hydrodynamic fluid film bearings

2.1.5 Hybrid Fluid Film Bearings

Hybrid fluid film bearings encompass any combination of the above mentioned technologies. Currently, configurations of interest are internally/externally fed hydrostatic/foil bearings. A schematic of a hybrid hydrostatic/foil bearing is shown in Figure 2-6. This figure shows an internally fed hydrostatic bearing in combination with a tension foil bearing. Measurements required to validate analytical models for hybrid bearings would be the same as the individual bearing technologies.

2.1.6 Hybrid (And) Magnetic Bearings

Magnetic bearings are not classified as fluid film bearings. However, they hold great potential and any significant bearing research program should consider planning for their inclusion. A schematic of a magnetic bearing is shown in Figure 2-7. There are basically two types of magnetic bearing systems: permanent magnet bias and all electromagnetic. Guidelines for magnetic bearing envelope requirements (courtesy of AVCON) are shown in Figure 2-8. Currently, foil bearings are being investigated as possible touchdown bearings for magnetic bearing systems. Therefore, it is wholly appropriate to include them in this study. Measurement requires for hybrid magnetic/fluid film bearings will be the same a the individual technologies. Measurements required to validate analytical models for magnetic bearings are as follows:

- Flux
- Current
- Voltage
- Bearing and coolant temperature
- Shaft motion, eccentricity, and tilt
- Coolant flow rate and shaft speed
- Load capacity
- Torque
- Rotordynamic force and moment coefficients
- Transient lift-off/touchdown characteristics

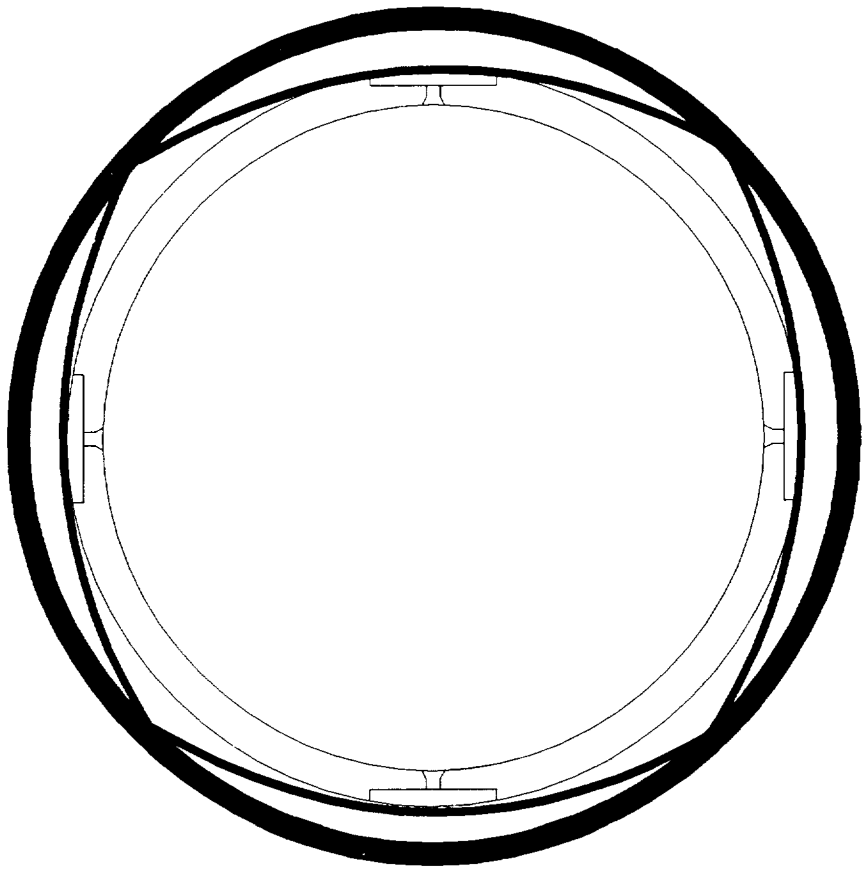


Figure 2-6. Hybrid hydrostatic/foil bearing

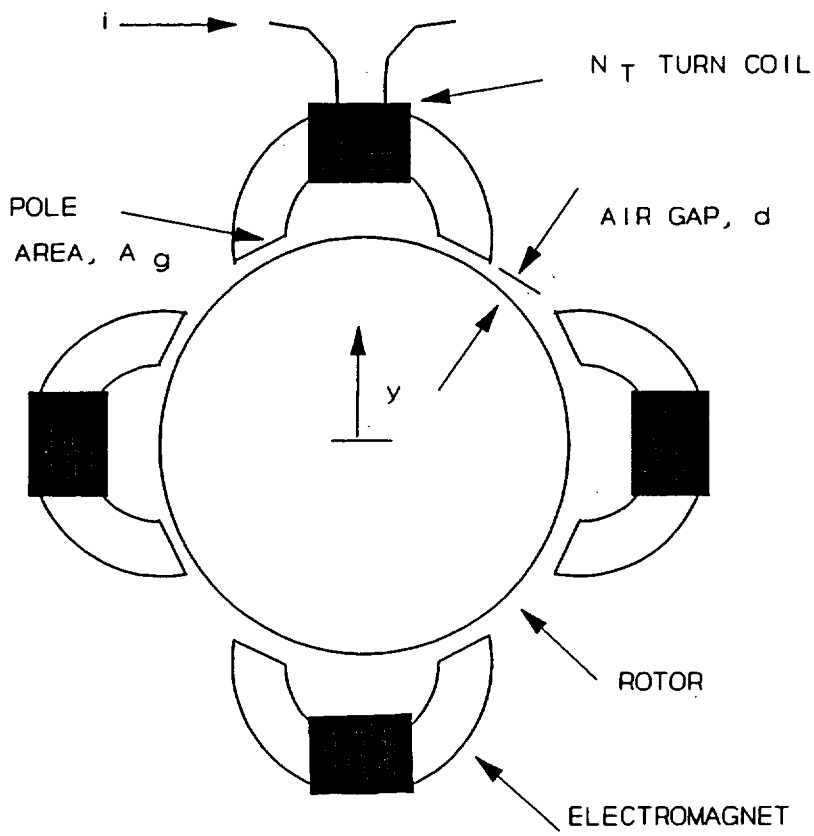
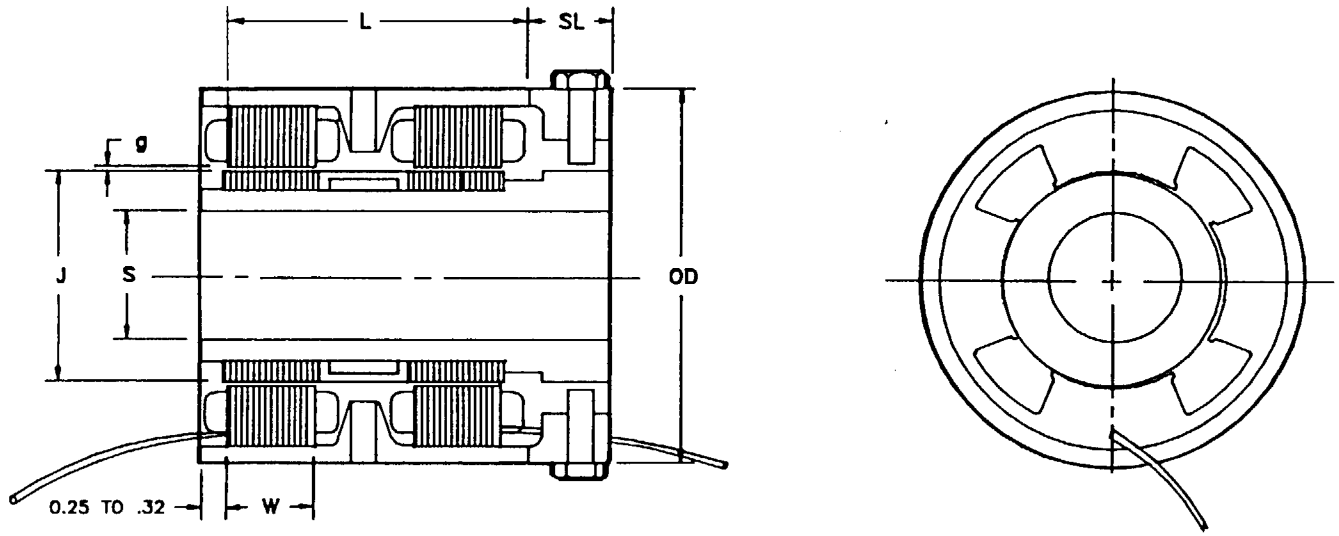


Figure 2-7. Magnetic bearing

FIGURE 3-3. AVCON MAGNETIC BEARING SIZES



AVCON MAGNETIC BEARING SIZES

STATIC LOAD CAPACITY (pounds)	S MAXIMUM SHAFT I.D.	J JOURNAL DIAMETER	OD OUTSIDE DIAMETER	L LENGTH OF BEARING	W LAMINATION WIDTH	SL SENSOR HSG.LENGTH	g AIR GAP	WEIGHT OF BEARING (pounds)	POWER** CONSUMPTION (watts)	MODEL NUMBER
2000	1.375	5.500	9.082	8.010	3.000	0.562	0.024	110.0	146	200055030
1000	1.28	4.000	6.750	5.630	2.060	0.562	0.024	42.8	106	100040056
500	1.23	3.000	5.200	3.800	1.371	0.562	0.020	16.7	65	050052038
400	1.22	2.750	4.800	3.360	1.195	0.562	0.020	12.5	59	040027534
300	1.37	2.750	4.645	2.500	0.898	0.562	0.018	8.4	49	030027525
200	1.375	2.500	4.210	1.900	0.658	0.562	0.018	5.25	43	020025019
125	1.087	1.750	3.295	1.800	0.585	0.562	0.015	3.1	28	012017518
50	0.928	1.187	2.600	1.125	0.343	0.562	0.015	1.3	20	005011812

Figure 2-8. Magnetic bearing envelope requirements

2.1.6 Thrust Bearings

The study of fluid film bearings for turbopump applications would not be complete if thrust bearings were not considered. The reaction of large transient thrust loads experienced during turbopump start-up has historically been accomplished using rolling element bearings. In order to realize the full potential of fluid film journal bearings, rolling element bearings must be completely eliminated from the design. Therefore, new technologies must be developed to replace them. There are many concepts for thrust bearings to be evaluated such as hard self lubricating rub washers, prepressurized hydrostatic, foil, magnetic, and centering ball bearings. Measurements required to validate analytical models for thrust bearings are as varied as the concepts themselves and would be similar to those required for journal bearings of the same technology.

2.2 TEST FLUIDS

The primary focus of this study will be on oxygen, hydrogen, and nitrogen in the liquid, gaseous, and mixed flow states. By limiting the study to these fluids the discriminating factors become the ability to function over a wide range of pressures and temperatures and LOX compatibility. A discussion of LOX compatibility issues for the current tester materials and specifications for current NASA Lewis facilities are given below. The use of air or hot gases is of secondary interest. Surrogate fluids such as HFC-134a and other CFC replacement fluids are not considered.

2.2.1 LOX Compatibility

Compatibility of the current tester materials with LOX is a major safety and cost concern. Absolute compatibility with LOX is difficult to achieve, however, methods have been developed to judge materials relative to each other. Table 2-1 shows a comparison of material compatibility parameters for the current materials for each tester.

Table 2-1. LOX compatibility factors for tester materials

	Max. Self Extinguishing Pressure (psi)				Ignition PV
Material	1/8 in dia	3/16 in dia	1/4 in dia	3/8 in dia	W/m^2 X10^{-8}
Inconel 718	500	500	500	1000	1.1-1.2
316 SS	500	500	1000	3000	0.5-0.7
Invar	<500	--	--	--	0.6-0.7

NOTE: Data courtesy of NASA White Sands Test Facility

2.2.2 NASA Test Facility CCL Cell 1

This is a recirculating liquid oxygen (or liquid nitrogen) facility which consists of a 3000 gallon, 250 psig supply tank feeding either one of two LOX pumps to supply the test rig. From the test rig, the LOX is throttled back down to near ambient pressure and returned to the supply tank which is vented to atmosphere. The first LOX pump is a reciprocating pump which can provide 1500 psig LOX at up to 75 GPM. The second LOX pump is a centrifugal pump which can provide 750 psig at up to 50 GPM. Flow to and from the test rig is controlled using hydraulically operated cryogenic valves operated from a remote control room located approximately 200 feet from the test cell. Gaseous nitrogen (2400 psig, 3 lbm/s, 250,000 scfm), gaseous helium (2400 psig, 0.5 lbm/s, 70,000 scfm), and service air (120 psig, 2 lbm/s, and unlimited volume) are also available.

2.2.3 NASA Test Facility CCL Cell 2

This is a blow down liquid hydrogen (liquid nitrogen) facility which consists of a 1300 gallon, 1440 psig run tank supplied by a 12000 gallon, 50 psi dewar. The run tank is pressurized using two 70,000 scf tube trailers. A vacuum jacketed line runs into the test cell from the run tank and can provide up to 2 lbm/s to the test rig. The hydrogen is then vented to a burnoff stack. Flow to and from the test rig is controlled using hydraulically operated cryogenic valves operated from a remote control room located approximately 300 feet from the test cell. Gaseous nitrogen (2400 psig, 3 lbm/s, 250,000 scfm), gaseous helium (2400 psig, 0.5 lbm/s, 70,000 scfm), gaseous hydrogen (2400 psig, 2 lbm/s, 140,000 scf), and service air (120 psig, 2 lbm/s, and unlimited volume) are also available.

2.3 INSTRUMENTATION

The ability to accommodate a large number and wide variety of instruments is a very important feature since NASA's objective is to develop a facility which can produce data for the validation of analytical predictions for fluid film bearing performance. Instrumentation of interest include:

- Pressure transducers
- Temperature transducers
- Accelerometers
- Force transducers
- Torque measurement devices
- Proximity probes
- Slip rings and transmitters for all of the above

2.3.1 Pressure Measurement

Standard pressure measurements such as supply and sump values must be made for all bearing configurations. For hydrostatic bearings, pressure measurements close to the orifice inlet will enhance the value of the test data. Since the primary function of a fluid film bearing is load support, determination of the load and verification of the analytical model are paramount. Load capacity in a fluid film bearing is determined from the pressure profile internal to the bearing. For hydrostatic bearings this requires measurement in the recess and land areas. Internally fed hydrostatic bearings require measurement of pressures at the shaft centerline and at the orifice inlet. Annular seals require pressure measurements along the length. Foil and other hydrodynamic bearings require extensive mapping of the pressure field internal to the bearing in order to identify regions of cavitation and maximum pressure.

2.3.2 Temperature Measurement

Standard temperature measurements such as supply and sump values must be made for all bearing configurations. Internal bearing temperatures are as important as pressure measurements for determining bearing characteristics. The characteristics of hydrodynamic bearings are especially dependent on fluid density and viscosity which are sensitive to temperature variations. In some instances, bearing stator material temperature profiles may be required to determine heat transfer characteristics for real properties and multi-phase flow analyses.

2.3.3 Acceleration Measurement

The measurement of acceleration is usually accomplished by locating accelerometers on the outer housing. For high speed, cryogenic fluid testers such as the HBT and OTV, accelerometer measurements may be the only means by which to determine the shaft speed. Eddy current proximity probes have relatively low frequency response limits and fiber optic probes are sensitive to bubbles and fluid density variations. Internal tester acceleration measurements are only necessary when testing for rotordynamic coefficients using a free floating test article.

2.3.4 Force Measurement

Force measurement is important for determining both static load and rotordynamic coefficients. For this study, the extraction of rotordynamic coefficients will be considered but is not essential.

2.3.5 Torque Measurement

Torque measurement is one the most difficult to make for individual bearings in a tester. The problem is magnified in this case by the use of turbine drives on all candidate testers. With electric motor drive systems, motor current can be used as well as telemetry systems mounted on the coupling to determine shaft torque. The ability to measure torque using any means possible will be discussed. However, it is not expected to be the discriminating factor.

2.3.6 Proximity Probes

The measurement of shaft location and dynamic response is every bit as important as pressure measurements for the validation of analytical models. Relative shaft/bearing measurements should be made in multiple planes with four probes in each plane. For the measurement of tilt effects, a minimum of two planes of probes per bearing is required. Proximity probes are also effective shaft speed sensors. For all applications, the probes (both eddy current and fiber optic) should reside in a stable pressure and temperature environment.

2.3.7 Slip Rings And Transmitters

The need for slip rings and/or transmitters is dependent on the desire to test internally fed hydrostatic bearings or to measure shaft strain as a means of determining loads. For high speed testing, slip ring assemblies may require an extensive amount of coolant hardware.

2.4 TEST OBJECTIVES

The requirements for fluids and instrumentation are dependent on the objective of the test. The three major bearing testing issues are:

- Performance
- Dynamic Characteristics
- Life

Bearing performance encompasses the measurement of both internal and external characteristics. Detailed flow field measurements must be made for complete validation of analytical models. Measurement of macro quantities such as leakage, torque, and load capacity are needed to establish the applicability of the bearing technology to different applications and for validation of analytical models.

The determination of dynamic characteristics can also be accomplished in different ways. The tester can be configured with simulated rotor mass, rotor inertia, pressure, temperature, and shaft speed of a turbopump and dynamic characteristics of the system measured and compared to analytical predictions for system response. Another approach is to heavily instrument a single bearing and determine its stiffness, damping, and added mass characteristics directly.

The operating life of a turbopump bearing is short compared to industrial bearing applications. Therefore, the focus of bearing life testing is on the start and shutdown transients which result in large bearing loads and chaotic shaft motion. An exception is the possibility of long idle mode operation for nuclear thermal propulsion.

2.4.1 Flow Field Measurement

Bearing flow field measurement has the objective of defining the detailed pressure, temperature, and velocity profiles in the bearing bore. As a minimum, pressure and temperature measurements are required to validate even the most simplistic analytical model of bearing performance. Today, detailed measurement of the velocity profile is only feasible for large scaled models of bearings using laser based measurement equipment. For this study the use of laser based measurement techniques will not be considered.

2.4.2 Bearing Performance Measurement

Bearing performance information is needed to establish the basic characteristics of the bearing with respect to load capacity, power consumption, and leakage. These are the most fundamental of test objectives and are the minimum criteria for acceptability.

2.4.3 Simulation Of Rotordynamic Characteristics

The most basic of dynamic tests is the rotordynamic simulation. The objective of this test is to measure the system dynamic response: critical speeds, forced response, and sub/supersynchronous frequencies. These responses can be compared to analytical predictions resulting in a basic understanding of how particular bearing configurations perform relative to one another.

The critical speed is defined as the speed of the shaft at which the response of the system is a maximum. Every rotating machine has multiple critical speeds which occur at natural frequencies which are related to bearing and shaft characteristics. A sample critical map for a fluid film bearing supported rotor is shown in Figure 2-9. The figure shows two closely spaced peak responses at low frequency which are generally related to bearing stiffness, the amplitude of the response is generally limited by system damping. A comparison of experimental and theoretical critical speed maps will yield information on effective bearing stiffness and damping.

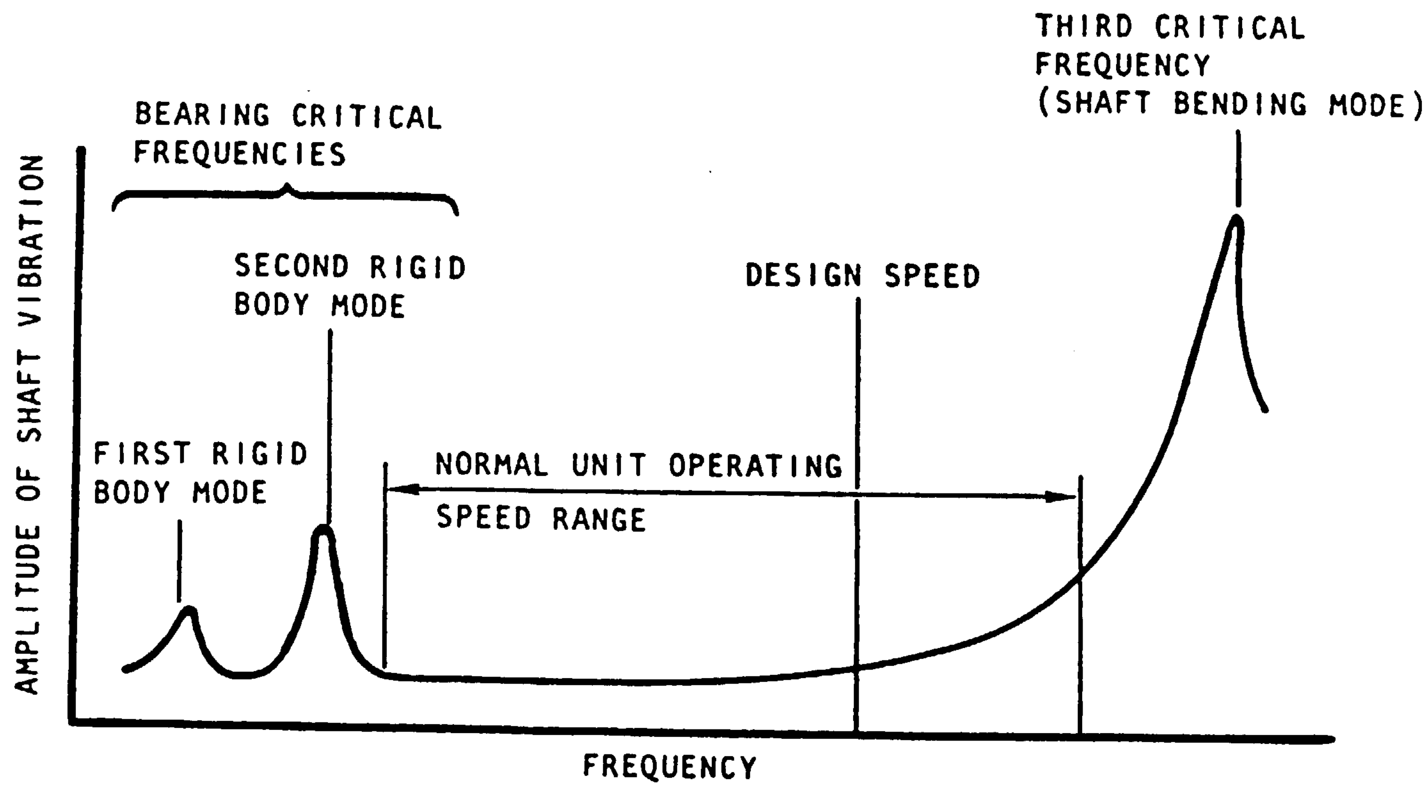


Figure 2-9. Sample critical speed map

The forced response of a turbomachine is the amplitude of shaft motion which is caused by residual unbalance of the shaft. This unbalance occurs to some degree in every rotating shaft. The degree of unbalance can be changed using weights added to the shaft at various axial locations. The change in response due to change in unbalance yields valuable information on bearing dynamic characteristics. Sample forced response predictions are shown in Figure 2-10. The figure show the deflected shape of the shaft at various axial locations and orbit shapes. A comparison between experimental and theoretical shaft orbits will yield information regarding bearing stiffness asymmetry.

Generally sub/supersynchronous frequencies which are related to destabilizing mechanisms present in seals, bearings, impellers, etc. are identified using experimentally obtained waterfall plots. A sample waterfall plot for a fluid film bearing support system is shown in Figure 2-11. The figure shows shaft responses at multiple frequencies. The dominant frequency is the synchronous speed of the shaft and the response is caused by the aforementioned unbalance. The smaller response at the lower frequency is caused by fluid film bearing forces related to cross-coupled stiffness. At some point, the shaft speed will reach a value it cannot exceed without experiencing violent unstable motion. A comparison of experimental and theoretical results for this onset speed of instability (OSI) yields valuable information regarding bearing cross-coupled stiffness and direct damping.

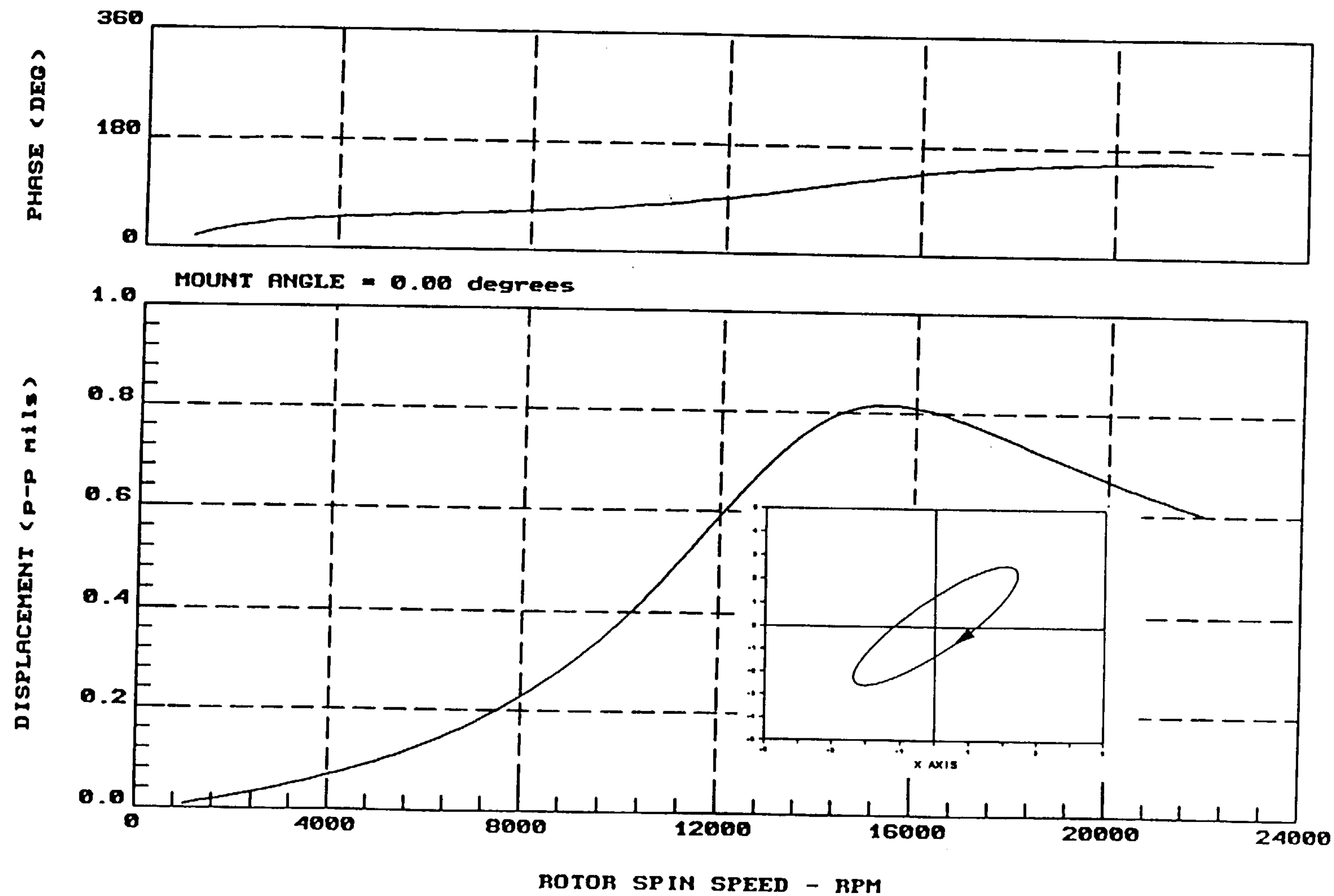


Figure 2-10. Sample unbalance response calculations

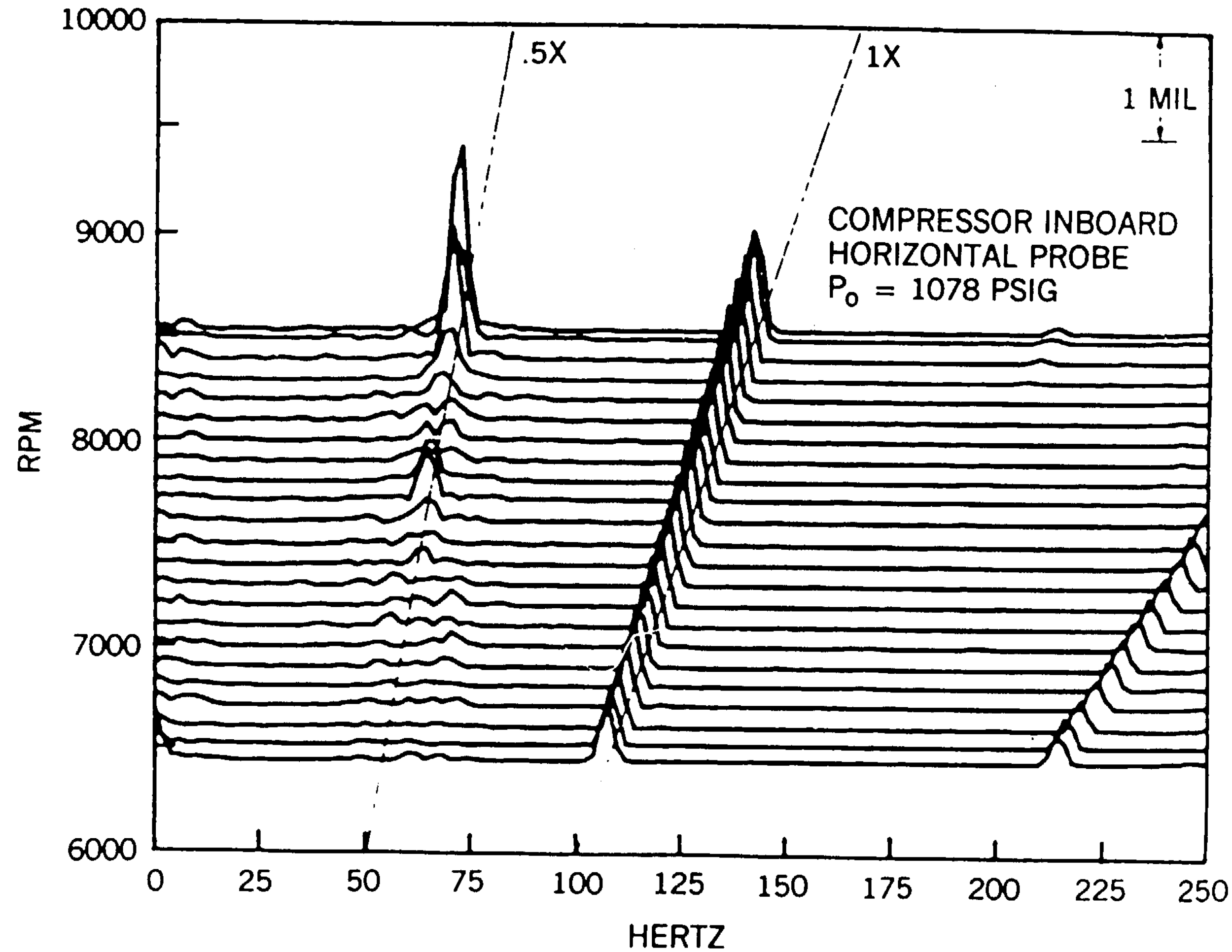


Figure 2-11. Sample waterfall plot

2.4.4 Extraction Of Rotordynamic Coefficients

The determination of rotordynamic coefficients is important for both the development of new bearing technologies, validation of analytical models, and the application of bearing technology to high speed rotating machinery. While fluid film bearings hold great potential for long life, low cost, and reliable operation, there is a general dearth of experimental data for their rotordynamic performance. This is especially true of foil bearings and other bearings operating in cryogens.

For rotordynamics, bearing testing has had the objective of determining the coefficients in the following bearing force-displacement model:

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} \\ K_{yx} & K_{yy} \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} \\ C_{yx} & C_{yy} \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \end{Bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} \\ M_{yx} & M_{yy} \end{bmatrix} \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \end{Bmatrix} \qquad (1)$$

where $(X,Y,\dot{X},\dot{Y},\ddot{X},\ddot{Y})$ are the rotor displacements, velocities and accelerations. (F_x,F_y) are the fluid film forces acting on the rotor (as illustrated in Figure 2-12), and (K,C,M) are the stiffness, damping, and inertia coefficients of the fluid film, respectively. For bearings with a large length to diameter ratio a more extensive force-displacement model may be required which takes into consideration angular rotation of the shaft. This force/moment-displacement/rotation model is expressed in the following form:

$$-\begin{Bmatrix} F_x \\ F_y \\ M_x \\ M_y \end{Bmatrix} = \begin{bmatrix} K_{xx} & K_{xy} & K_{x\alpha} & K_{x\beta} \\ K_{yx} & K_{yy} & K_{y\alpha} & K_{y\beta} \\ K_{\alpha x} & K_{\alpha y} & K_{\alpha\alpha} & K_{\alpha\beta} \\ K_{\beta\alpha} & K_{\beta y} & K_{\beta\alpha} & K_{\beta\beta} \end{bmatrix} \begin{Bmatrix} X \\ Y \\ \alpha \\ \beta \end{Bmatrix} + \begin{bmatrix} C_{xx} & C_{xy} & C_{x\alpha} & C_{x\beta} \\ C_{yx} & C_{yy} & C_{y\alpha} & C_{y\beta} \\ C_{\alpha x} & C_{\alpha y} & C_{\alpha\alpha} & C_{\alpha\beta} \\ C_{\beta x} & C_{\beta y} & C_{\beta\alpha} & C_{\beta\beta} \end{bmatrix} \begin{Bmatrix} \dot{X} \\ \dot{Y} \\ \dot{\alpha} \\ \dot{\beta} \end{Bmatrix} + \begin{bmatrix} M_{xx} & M_{xy} & M_{x\alpha} & M_{x\beta} \\ M_{yx} & M_{yy} & M_{y\alpha} & M_{y\beta} \\ M_{\alpha x} & M_{\alpha y} & M_{\alpha\alpha} & M_{\alpha\beta} \\ M_{\beta x} & M_{\beta y} & M_{\beta\alpha} & M_{\beta\beta} \end{bmatrix} \begin{Bmatrix} \ddot{X} \\ \ddot{Y} \\ \ddot{\alpha} \\ \ddot{\beta} \end{Bmatrix} \qquad (2)$$

The aforementioned force-displacement models of Equations 1 and 2 are linear models. The term linear refers to the inherent assumption that the force coefficients K,C, and M are not frequency dependent. It is not good practice to assume the characteristics of data ahead of time. In some cases, such as gas hydrostatic bearings and magnetic bearings, analysis will indicate ahead of time that the linear model may not apply. Therefore, the actual measurement objective for dynamic testing of bearings (and also seals) is the transfer function. Equation 1 can be rewritten in terms of the complex transfer function as follows:

$$-\begin{Bmatrix} F_x \\ F_y \end{Bmatrix} = \begin{bmatrix} H_{xx} & H_{xy} \\ H_{yx} & H_{yy} \end{bmatrix} \begin{Bmatrix} X \\ Y \end{Bmatrix} \qquad (3)$$

where (for the linear model of Equation 1)

$$H_{ij} = K_{ij} - \omega^2 M_{ij} + j\omega C_{ij}$$

Once obtained from test data, the transfer functions will determine whether or not the linear model applies. If the linear model does apply, the K,C,M coefficients can be obtained using linear and polynomial curve fits of the transfer function data. If the linear model does not apply, the transfer function information is used in the rotordynamic analysis in place of stiffness, damping, and mass coefficients. An example set of transfer functions for fluid film bearing testing is shown in Figure 2-13. A similar expression can be developed for the more complicated model of Equation 2.

Forces Developed in a Fluid Film Element

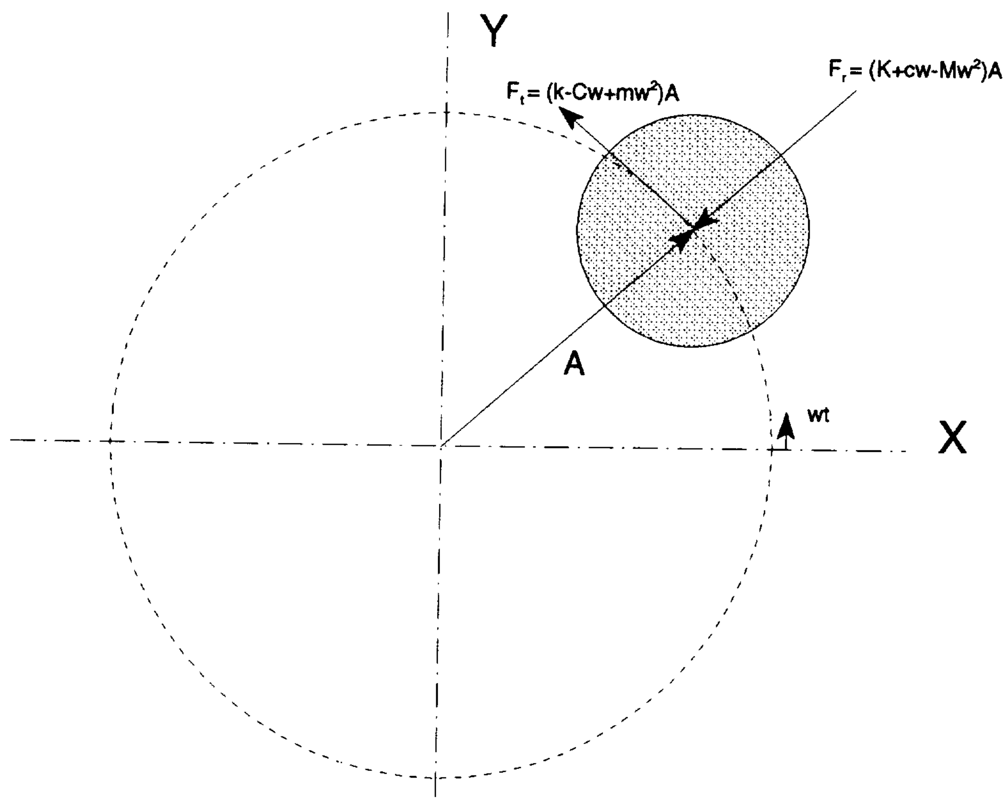


Figure 2-12. Fluid film forces acting on a shaft

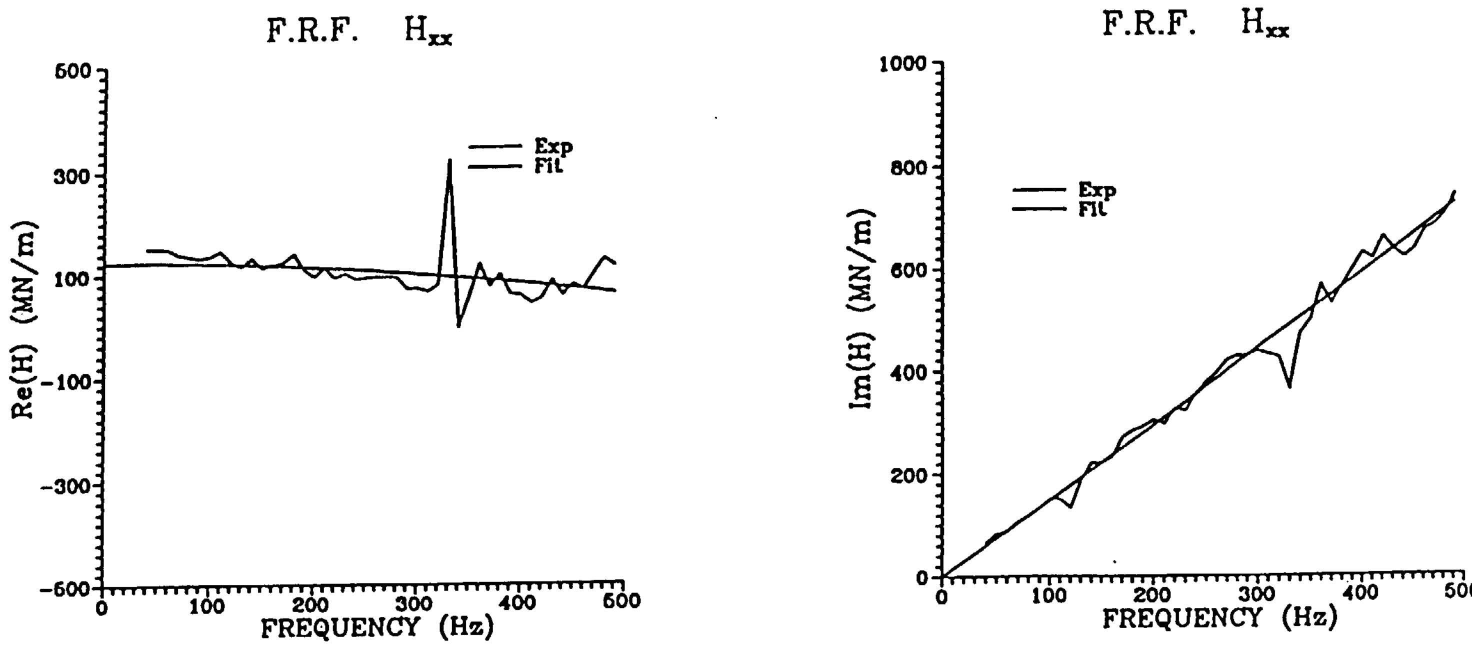


Figure 2-13. Sample transfer function data from fluid film bearing test

2.4.5 Transient Start/Shutdown Simulation

One of the major issues retarding acceptance of fluid film bearings for turbopump applications is the potential for rubbing, and the resulting wear during start and shutdown transients. Turbopump start-ups are characteristically violent with rapid shaft acceleration. A sample SSME HPOTP start transient is shown in Figure 2-14. The figure shows that the shaft accelerates at a rate of up to 10,000 rpm/s over portions of the speed profile. Fixed loads on the bearings during this start transient are shown in Figure 2-15. With the proper facility, a tester with accurate speed and radial load capability can accurately simulate this and other start/shutdown transient profiles. Successful determination of start transient characteristics depends on the available instrumentation as well as facility capability.

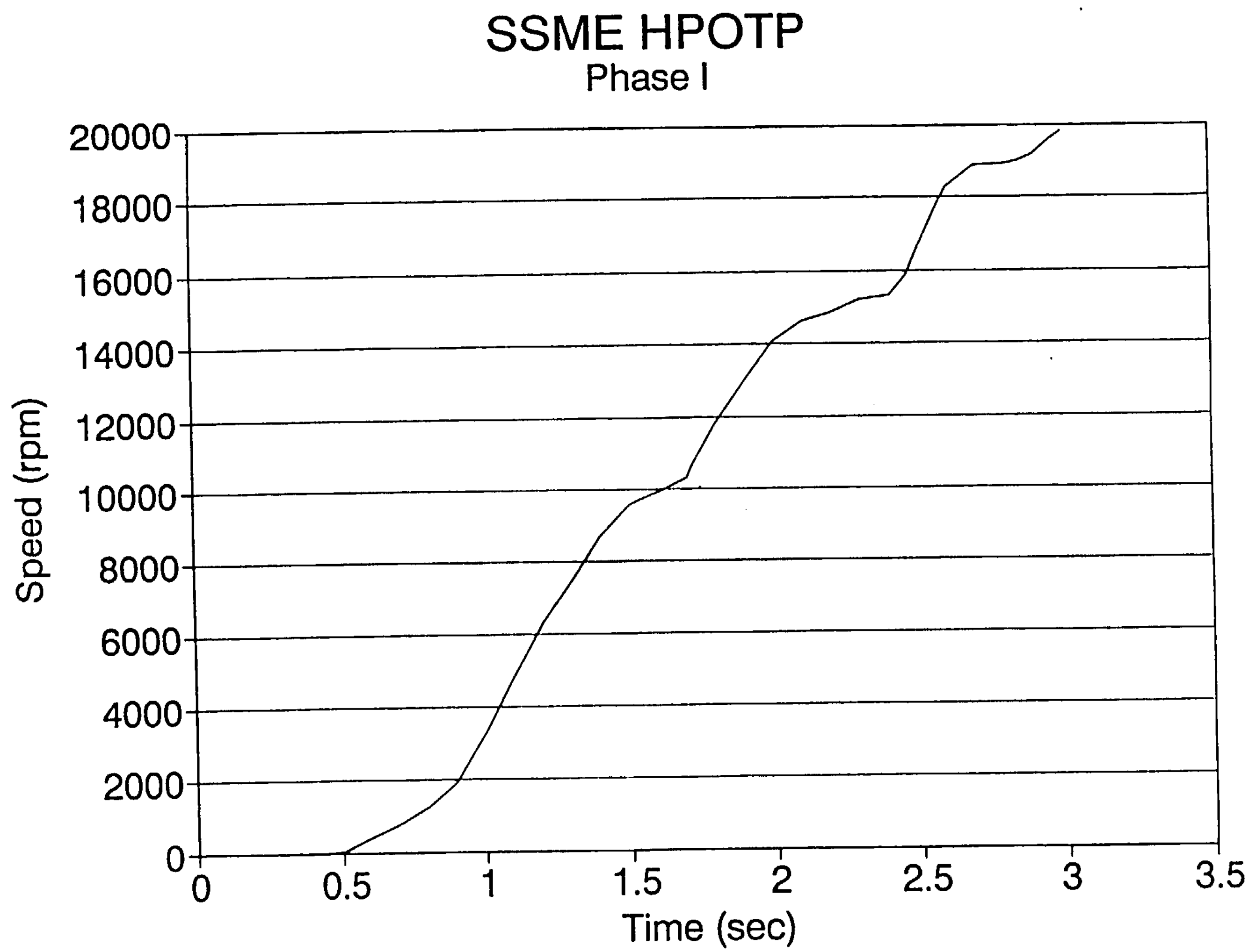


Figure 2-14. Start transient speed profile of SSME HPOTP

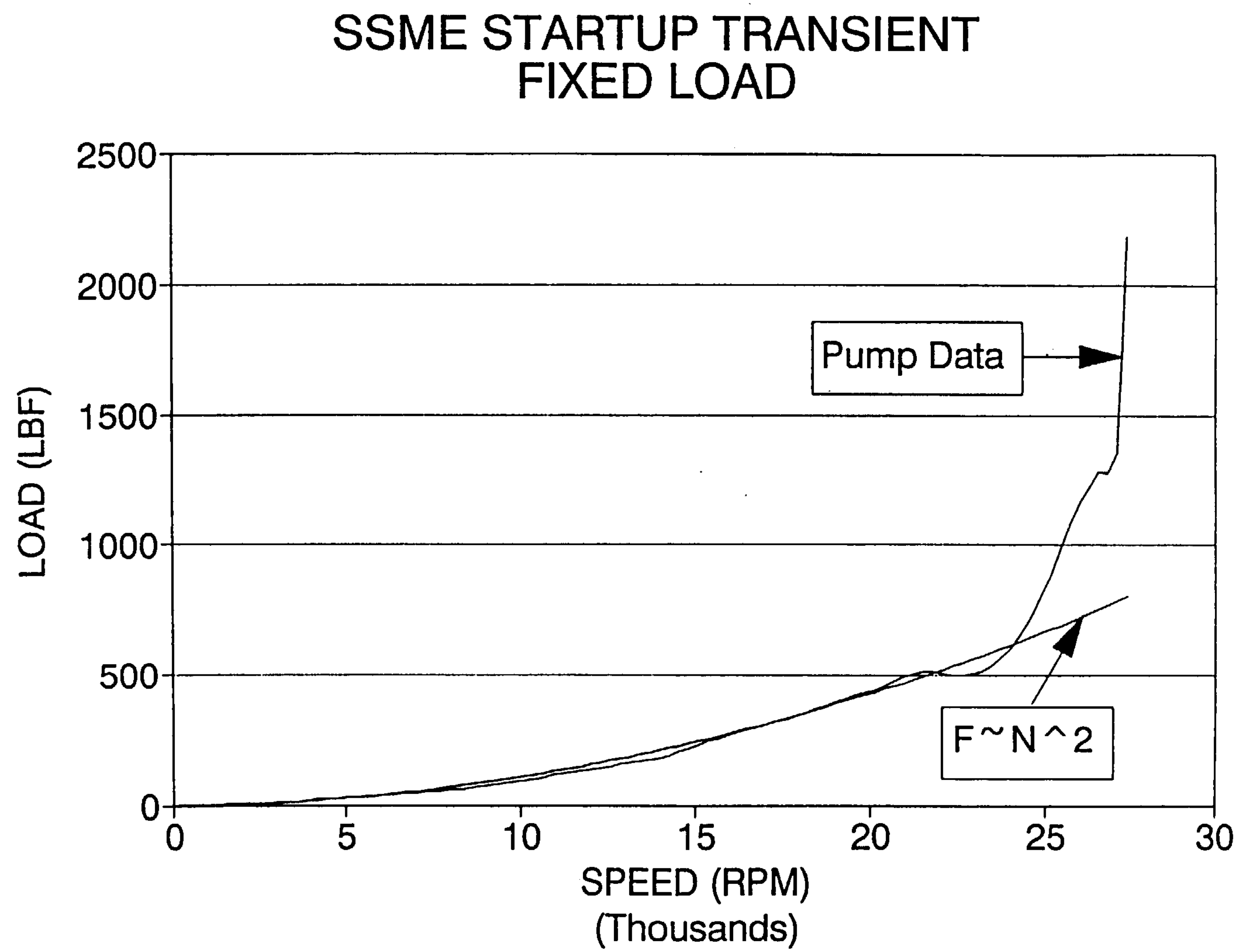


Figure 2-15. Start transient radial load profile of SSME HPOTP P/E bearing

3.0 TECHNICAL EVALUATION

3.1 HYBRID BEARING TESTER

The Hybrid Bearing Tester was originally designed for liquid hydrogen testing and has the following capabilities:

- 200 lbf. radial load capability (100 lbf./bearing)
- 650 psi supply pressure
- 300 psi sump pressure
- 120,000 rpm, 16.2 Hp turbine drive

3.1.1 Test Article Configurations

The following discussions focus on testing identical bearings at each support bearing location. However, the large radial loader cavity in this tester lends itself to accommodating a test bearing which may be different than the support bearings. This could be an attractive alternative for bearings requiring a large envelope, such as the magnetic bearing.

3.1.1.1 Hydrostatic Bearings. The tester was originally designed for externally fed hydrostatic bearings. Therefore, the basic construction is well suited to their testing. However, key limitations of the tester are the low supply pressure capability (650 psi) and the restricted bearing diameter due to the presence of the loader cavity ring seals. It is unlikely that one would design a pump to operate on hydrostatic bearings with only a 650 psi supply pressure at speeds approaching 100,000 rpm. If the ring seals were eliminated, practical hydrostatic bearing diameters would be in the range 20-60 mm. Bearing length is limited to 30 mm. Bearing L/D could be varied between 0.0 and 1.5, which is the range of interest for most hydrostatic bearing applications.

For internally fed hydrostatic bearings, the tester has several limitations; access for supply fluid, low thrust bearing capacity, and low bearing supply pressure. The supply flow for internal feed could be fed into the free end of the shaft using shaft seals and an extended shaft length (as shown in Figure 3-1). This would deliver the maximum supply pressure to the orifice inlet. However, the resulting thrust load would be much greater than the thrust bearing and turbine back pressure would be capable of reacting, for normal operation. There is also a danger related to the strength of the end plate bolts. The current sump pressure limit is 300 psi. A seal failure would expose the cavity to pressures as high as 650 psi which may lead to failure. The alternative is to remove the radial loader and feed the bearings through slots in the shaft (as shown in Figure 3-2). The main drawback in using this method is the large pressure loss at the entrance to the slots. This loss combined with the already low supply pressure capability would not result in a credible test. Another concern with respect to this method of supplying pressure is the need for the ring

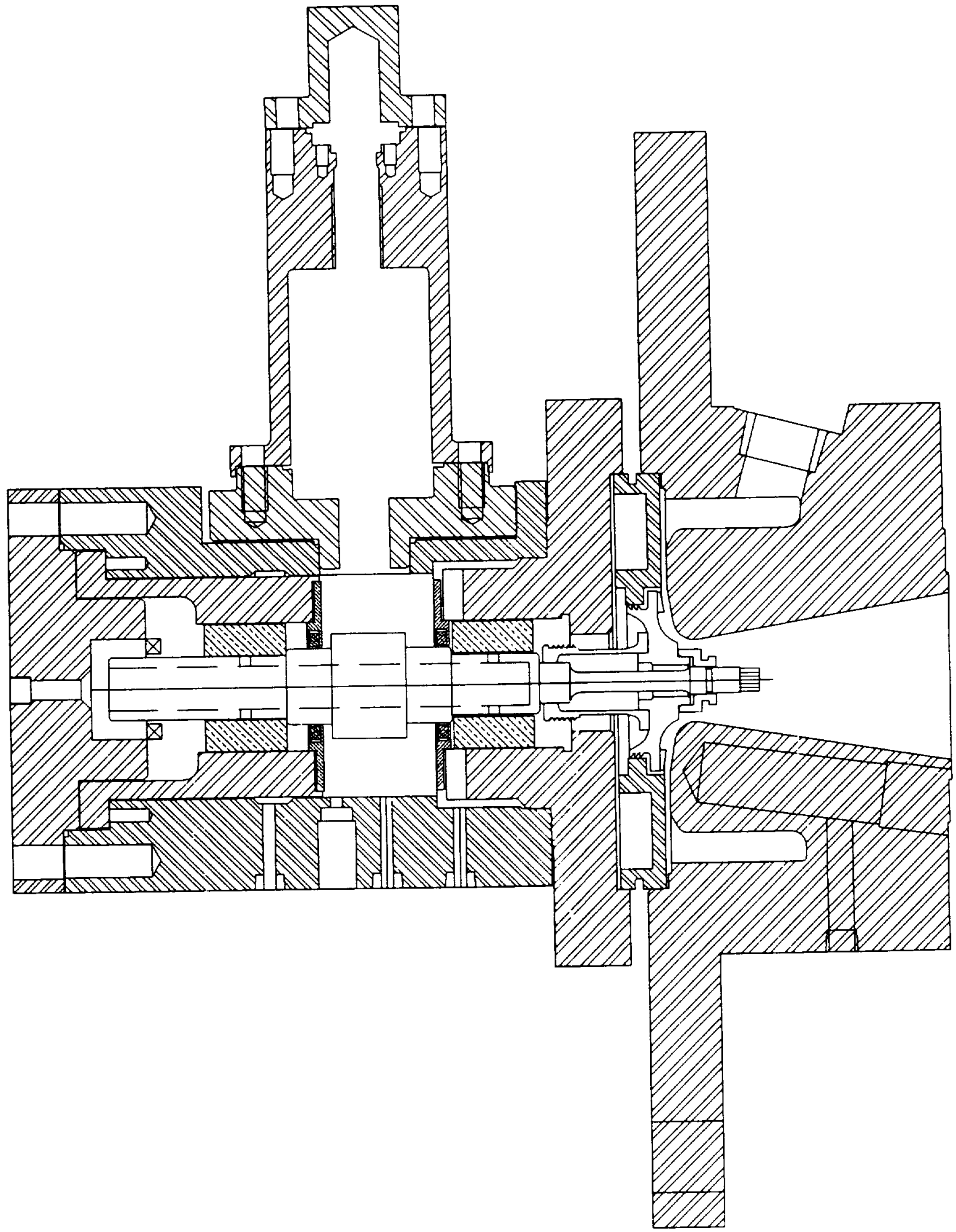


Figure 3-1. Fluid supply concept for internally fed bearing using shaft end bore

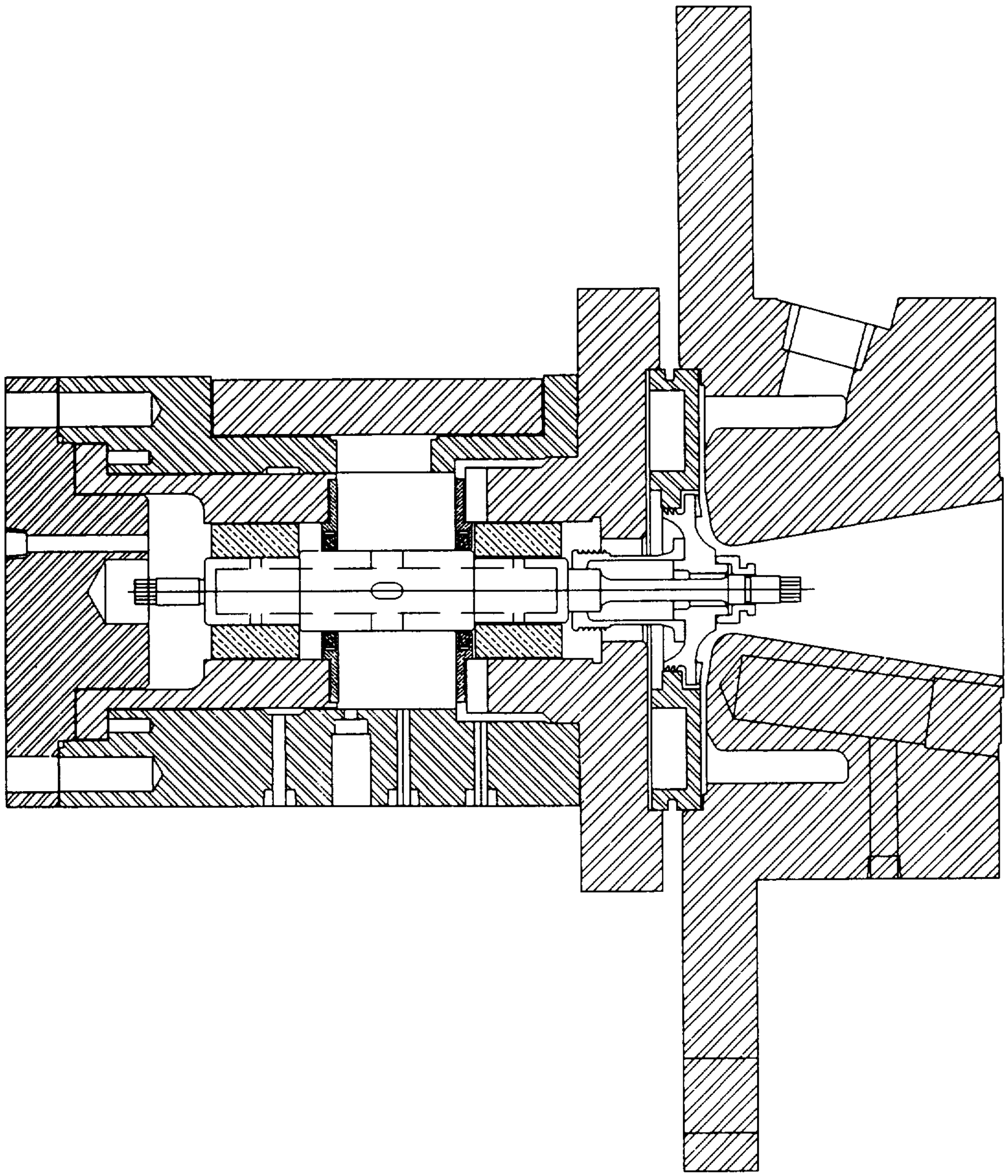


Figure 3-2. Fluid supply concept for internally fed bearing using center slot

seals which will limit the maximum diameter of the test article. The seals may also have to be redesigned to withstand the increased pressure.

3.1.1.2 Annular Seals. The tester is not well suited to the testing of annular seals primarily due to the lack of inlet flow control and measurement capability. Annular seal testing requires good control of both supply pressure and inlet velocity vector. Accurate measurement of the inlet velocity vector is necessary for the validation of any computer code with respect to rotordynamic coefficients or rotordynamic performance. The construction of the test apparatus precludes the use of conventional fluid velocity measurement devices such as pitot probes.

3.1.1.3 Foil Bearings. The flexibility in the bearing carrier design would allow either type of foil bearing to be tested. Without the ring seals, an L/D of 0.0-1.5 would be sufficient for typical bending type foil bearings and the available radial space is sufficient for tension type foil bearings for 20-30 mm bore sizes. Since foil bearing load capacity is characteristically low (50-250 psi) compared to other fluid film bearings, the 100 lbf. radial load capability of tester should be sufficient.

3.1.1.4 Hydrodynamic Bearings. Fixed geometry hydrodynamic bearings of all types should easily fit into this tester. However, the available envelope is not sufficient to accommodate bearings needing large radial space such as tilt pad bearings. Another concern is the limited load range of the loader. Typical hydrodynamic bearing load capacity is in the range 200-400 psi. Therefore, an applied load of 100 lbf. may not suffice to exercise the bearing in the desired eccentricity range.

3.1.1.5 Hybrid Fluid Film Bearings. The available bearing envelope of 20-60 mm diameter and 30 mm length should be sufficient to accommodate some, not all, of the currently proposed hybrid fluid film bearing concepts. Bearings such as the tilting pad hydrostatic bearing proposed by Aerojet for the NLS program will not fit.

3.1.1.6 Hybrid Magnetic Bearings. Referring to the information provided in Figure 2-8, it appears that it would be difficult to accommodate even the smallest and most efficient magnetic bearing design in the support bearing location without modification to the main tester housing. However, a magnetic bearing could be easily installed in the radial loader cavity. The high speed of the tester would also require laminations on the shaft which would require a new shaft design. A magnetic bearing which could be accommodated in the tester would have a load capacity of approximately 50-100 lbf. which is well within the 100 lbf. capability of the radial loader. The possibility of testing a hybrid bearing which includes a magnetic bearing (such as a foil/magnetic bearing combination) is not likely for the support bearing location considering the constraints already imposed by the magnetic bearing alone. However, a hybrid bearing could be installed in the radial loader cavity.

3.1.1.7 Thrust Bearings. This tester currently has a hydrostatic thrust bearing on the turbine end. This area could accommodate other thrust bearing concepts. The free end cover plate gives ample access to the end of the shaft so that other thrust bearing concepts could be installed with very few design changes. Thrust load can be varied using the back pressure of the turbine or by enhancing the turbine end thrust bearing for free end thrust bearing loading.

3.1.1.8 Test Article Summary. The discussions presented above reveal that within the envelope afforded at the support bearing after removal of the ring seals that most bearing configurations can be accommodated. Bearings requiring larger envelopes, such as the magnetic bearing and its hybrids, will not fit at the support bearing locations but can be accommodated in the radial loader cavity. Thrust bearing testing can be accomplished using a variety of design approaches and loading schemes.

3.1.2 Test Fluids

Since the tester was designed for liquid hydrogen use, test fluid issues are limited to LOX compatibility and use at higher temperatures. LOX compatibility can be assessed using the data provided in Table 2-1. The tester shaft is made of Inconel 718 which has good LOX compatible properties. The housing is made of 316 stainless steel which is a grade of steel which has marginal LOX compatible properties. Since all of the housing parts are made of the same materials, it is likely that the interference fits would remain somewhat constant at the slightly higher temperatures of LOX and LN2. However, no stress analysis information is given in the original MTI design report. Therefore, care must be exercised prior to testing in much hotter environments such as gaseous nitrogen or air. This tester has already shown proven ability to control cryogenic fluid conditions to provide multi-phase flow in the test article.

Based on the data provided in Table 2-1, it is not recommended that this tester be used in LOX. The housing material should be changed to Inconel 718. This would accomplish two goals: LOX compatibility and thermal coefficient of expansion consistent with the shaft material.

3.1.3 Instrumentation

The apparent ability of this tester to easily accept a wide variety and large number of measurement devices is one of its major advantages. The support bearings are completely contained in removable and changeable carriers. The radial loader cavity is spacious and can accommodate multiple instrumentation access ports. This flexibility would allow the configuration of a set of bearing carriers for any type of bearing and its associated instrumentation. Similar flexibility could be achieved for thrust bearing testing if free end bearings were used. Access to the turbine end thrust bearing is limited.

3.1.3.1 Pressure Measurement. While access to the turbine end bearings is limited, there are many possibilities for installing instrumentation in the free end bearing cavity. Simple modifications to the cover plate and bearing carrier could provide for complete arrays of pressure measurement ports which could be used to map the pressure contours of fluid film bearings. Measurement of pressures for internally fed hydrostatic bearings is possible but difficult. This would most likely require redesign of shaft and cover plate to accept a slip ring assembly.

3.1.3.2 Temperature Measurement. Like pressure measurements, temperature measurements could be made easily on the free end bearing. Access to the turbine end bearing is possible, but difficult. The flexibility afforded by the free end cavity would allow new temperature measurement techniques to be used such as the thermocouple grid currently being developed at Dartmouth.

3.1.3.3 Acceleration Measurement. Plenty of surface area is available for housing acceleration measurements, both radially and axially. Internal to the tester accelerometer mounting would be limited to the axial face of the free end bearing and any surface of a bearing mounted in the radial loader cavity.

3.1.3.4 Force Measurement. The load path on the support bearings are not conducive to accurate force measurement. Therefore, it is not advisable that force measurement be attempted for the support bearings without major modifications to the existing design. However, a bearing located in the radial loader cavity would provide ample access for force transducers.

3.1.3.5 Torque Measurement. Torque measurement for the individual support bearings is unlikely. For a bearing located in the radial loader cavity, torque measurements could be made using a variety of methods.

3.1.3.6 Proximity Probes. The turbine end support bearing and thrust bearing have very limited access for proximity probes. The free end support bearing may support all desired proximity measurements. Again, the most access for proximity probes is the radial loader cavity.

3.1.3.7 Slip Rings And Transmitters. The shaft diameter and possible fluid supply hole in the shaft are too small to accommodate transmitters. A slip ring assembly could be configured to feed out of the free end of the shaft.

3.1.3.8 Instrumentation Summary. The Hybrid Bearing Tester can accept most instrumentation for the support bearings. For support bearings, the best access is available at the free end bearing location. Instrumentation which can not be installed at the support bearings can be used for bearings installed in the radial loader cavity. The tester design can be modified to accept a slip ring at the free end of the shaft enabling measurements internal to the shaft.

3.1.4 Test Objectives

The overall test objective is the validation of analytical models. To that end, the most critical information is the bearing clearance profile. For cryogenic testing this value is difficult to determine at chill conditions. Common practice is to calculate the clearance function. However, this reduces the research effort to a validation of analytical models with an analytical value. To avoid this problem, tester materials for housing and shaft should be the same so that assembly clearances are the same as chill clearances. The HBT has a 316 SS housing and an Inconel 718 shaft which would require calculated clearance functions.

3.1.4.1 Flow Field Measurement

Detailed mapping of the bearing pressure and temperature profiles in the bore is possible by heavily instrumenting the free end bearing or a bearing located in the radial loader cavity. Instrumentation leads could be routed easily through a modified cover plate. No velocity measurements in the bore are possible due to lack of radial access across the interfaces of the various housing parts. This limitation would severely limit the ability to test annular seals. It is not feasible to test annular seals located in the radial loader cavity or fed through the free end cover plate due to excessive axial thrust. The lack of radial access also eliminates the possibility of performing flow visualization tests. However, a special one piece housing could be constructed which would allow an observation window to be installed. However, this would be difficult to manufacture (thermal expansion problems) and use (frost) in a cryogenic environment.

Overall, this tester is capable of extensive flow field pressure and temperature mapping for the bearing configurations which it can accept.

3.1.4.2 Bearing Performance Measurement

The measurement of leakage is very straight forward for externally fed hydrostatic bearings and other bearings fed through internal housing annuli. Difficulties arise for internally fed hydrostatic bearings and other bearing configurations which rely on the ring seals for containment. Leakage measurements for these configurations will always have uncertainties of the same magnitude as the seal leakage. With so many common drains, the seal leakage can not be measured accurately, it must be calculated based on pressure measurements.

The determination of load capacity is fairly straightforward. The only problem with the tester is that the loader range is somewhat limited with respect to simulating loads for oxygen pumps. A more capable radial loader could be obtained by eliminating the hydrostatic load shoe in favor of a duplex ball bearing attachment.

Accurate measurement of torque for the support bearings is virtually impossible. Too many load paths exist by way of the thrust bearing, ring seals, and radial loader for an accurate assessment of individual bearing torque to be made. There is also large uncertainty in the power supplied by the turbine. If accurate torque is required, a heavily instrumented bearing located in the radial loader cavity would provide the best chance of success.

Overall, this tester is capable of meeting the requirements for bearing performance measurement. However, a successful test will require careful planning, proper facility/tester configuration, and extensive calibration.

3.1.4.3 Simulation Of Rotordynamic Characteristics. The Hybrid Bearing Tester is very well suited to performing rotordynamic simulations. The free end of the shaft provides access for an extra mass to simulate impeller weight and inertia, the radial loader can impart side loads and be enhanced to accommodate a larger or differently shaped shaft section, and the turbine wheel has a labyrinth seal which could have flow injected to impart destabilizing forces on the shaft. However, with these constraints the tester is limited to bearings which fit at the support locations. Larger bearings will require elimination of the ring seals. Extremely large bearings will have to be tested in the radial loader cavity for other test objectives can not be accommodated for rotordynamic simulation.

3.1.4.4 Extraction Of Rotordynamic Coefficients. The determination of rotordynamic coefficients is possible using this tester provided that the test article is placed in the radial loader cavity. Furthermore, extensive modification of the radial loader would be required to create a loader/impactor combination in addition to housing modifications to accept a second impact device. This impactor is better constructed with a ball bearing loader.

3.1.4.5 Transient Start/Shutdown Simulation. This tester has already been used successfully for transient start/shutdown simulation in liquid hydrogen. For the other fluids of interest (nitrogen and oxygen), the radial loader capacity would have to be increased to provide a more realistic load environment. This could be accomplished by replacing the hydrostatic shoe with a rolling element bearing and improving the strength of the loader housing and flange bolts. This tester is particularly attractive for this task because of the even loading of the support bearings which eliminates concerns regarding edge loading and tilt during the transient. Another key consideration for life testing is the ability to accept different materials. The modular nature of the housing allows for design flexibility when selecting a material for the bearing carrier to alleviate thermal expansion issues. For thrust bearing transient testing, the pressures in the turbine discharge and free end bearing cavities could be controlled sufficiently to provide the required load characteristics.

3.1.4.6 Test Objective Summary. The Hybrid Bearing Tester is capable of performing all of the required testing functions provided that design modifications are made. However, its size sets a practical limit for bearings configurations which can be tested. Redesign of the radial loader cavity would significantly enhance capability.

3.1.5 Hybrid Bearing Tester Summary

Based on the aforementioned discussion, it can be concluded that the Hybrid Bearing Tester is flexible enough to accomplish most of the test objectives for most of the bearings of interest and can accommodate a large amount of instrumentation. Size and configuration constraints restrict the tester from being used for all tests with all bearings. The major issues identified with respect to the evaluation criteria are LOX compatibility, material thermal expansion incompatibility, low supply pressure, radial loader capacity, and elimination of ring seals. However, these issues are related and can be addressed through design improvements and material changes. The only tasks for which this tester is not suitable are the testing of annular seals and rotordynamic and transient simulation of bearings requiring large housing envelopes.

3.2 OTV BEARING TESTER

The OTV Bearing Tester was originally design for liquid hydrogen testing and has the following capabilities:

- 30 lbf. radial load (15 lbf./bearing)
- 3000 psi supply pressure
- 300 psi sump pressure
- 200,000 rpm turbine drive

3.2.1 Test Article Configurations

Test article installation is limited in this tester to existing radial and thrust bearing locations. The radial loader is fixed and the solid one piece housing eliminates the possibility of modifying this cavity to accept other configurations.

3.2.1.1 Hydrostatic Bearings. The tester was originally designed for externally fed hydrostatic bearings. Therefore, the basic construction is well suited to their testing. The orifice pin configuration shown in the design report should be avoided to prevent possible leakage and pressure containment issues. Practical hydrostatic bearing diameter is limited to 25-30 mm. Critical housing features prevent enlargement of the housing to accept larger sizes. Thrust bearing and turbine geometry requirements restrict the ability to use a smaller shaft. Bearing length is limited to 25 mm. Therefore, bearing L/D could be varied between 0.0 and 1.0, which is the acceptable but limiting for most hydrostatic bearing applications.

The OTV Bearing Tester is not capable of testing internally fed hydrostatic bearings. The ends can not be used for fluid supply because of thrust control concerns and sufficient space and access is not available for modification of the shaft and housing for slot feed.

3.2.1.2 Annular Seals. The OTV Bearing Tester is not well suited to the testing of annular seals primarily due to the lack of inlet flow control and measurement capability. Annular seal testing requires good control of both supply pressure and inlet velocity vector. Inlet ports adjacent to the radial loader could be used for inlet flow control. However, there is not a sufficient number of ports to insure that uniformity of inlet flow is maintained. If annular seal testing is desired, a new housing would have to be designed and manufactured with the correct number of supply ports, inlet guide vanes, and access for the necessary instrumentation.

3.2.1.3 Foil Bearings. It would be difficult to test either configuration of foil bearing in this tester. The bending type of foil bearing has a sway space of about 0.02-0.03 in. The close clearance of the fixed geometry radial loader may lead to a rub condition when testing this type of bearing. Insertion of a tension foil bearing into the tester is limited by the available housing envelope. The basic design of the tester is not flexible enough to enable the testing of foil bearings even with substantial redesign.

3.2.1.4 Hydrodynamic Bearings. Fixed geometry hydrodynamic bearings of all types should easily fit into this tester. However, the available envelope is not sufficient to accommodate bearings needing radial space such as tilt pad bearings. Another concern is the limited load range of the loader. Typical hydrodynamic bearing load capacity is in the range 200-400 psi. Therefore, an applied load of 15 lbf. may not suffice to exercise the bearing in its desired eccentricity range.

3.2.1.5 Hybrid Fluid Film Bearings. The available bearing envelope of 25-30 mm diameter, 25 mm length, and restricted shaft size is probably not sufficient to accommodate most of the currently proposed hybrid fluid film bearing concepts.

3.2.1.6 Hybrid Magnetic Bearings. Referring to the information provided in Figure 2-8, it appears that it is impossible to accommodate even the smallest and most efficient magnetic bearing design. The high speed of the tester would also require laminations on the shaft which would require a new shaft design.

3.2.1.7 Thrust Bearings. This tester currently has hydrostatic thrust bearings at each end of the shaft. Different configurations of thrust bearings could be inserted into either location. However, there is no mechanism available to control the axial load, axial position of the shaft, or make useful measurements.

3.2.1.8 Test Article Summary. The discussions presented above reveal that within the envelope afforded by the bearing locations that very few of the conventional and possibly none of the hybrid fluid film bearings can be accommodated.

3.2.2 Test Fluids

Since the tester was designed for liquid hydrogen use, test fluid issues are limited to LOX compatibility and use at higher temperatures. LOX compatibility can be assessed using the data provided in Table 2-1. The tester shaft is made of titanium which is known to have extremely poor LOX characteristics and was not included in the NASA WSTF database. The housing is made of Invar which has marginal LOX compatible properties. Since all the housing is a single piece, thermal expansion issues at higher temperatures are limited to bearing inserts. The stress analysis provided in the original design report details the difficulties in accommodating alternate materials for the bearings. Therefore, substantial analysis will be required prior to testing in much hotter environments such as gaseous nitrogen or air. No information on test experience with this tester is available in the public domain.

Overall, this tester will be difficult to use with different bearing materials in liquid hydrogen and liquid nitrogen and is unacceptable for use in LOX.

3.2.3 Instrumentation

The fixed housing design of this tester severely limits the ability to accommodate large numbers and a wide variety of instrumentation.

3.2.3.1 Pressure Measurement. Access for pressure measurements are limited to a few radial inlet ports. The housing and bearing insert are so restricted by existing ports and retainers that access for new instrumentation is not available.

3.2.3.2 Temperature Measurement. Like pressure measurements, temperature measurements can not be made easily in this tester.

3.2.3.3 Acceleration Measurement. Plenty of surface area is available for housing acceleration measurements, both radially and axially. Acceleration measurements can not be made internal to the tester.

3.2.3.4 Force Measurement. No space is available for any type of force measurement instrumentation.

3.2.3.5 Torque Measurement. Torque measurement for the support bearings is unlikely. The housing can not be modified to accept an instrumented bearing carrier.

3.2.3.6 Proximity Probes. Proximity probes are limited to their existing locations. The probes on the turbine end are suspect because of the possibility of wide temperature variations as the pressure to the turbine inlet and therefore the gas velocity is increased. To function properly, the turbine end of the tester would have to be extended so that the proximity probes could be moved inside of the labyrinth seal.

3.2.3.7 Slip Rings And Transmitters. The tester will not accommodate slip rings and transmitters. Thrust control requirements eliminate access to shaft ends and the size of the shaft is too small to accept a transmitter.

3.2.3.8 Instrumentation Summary. The OTV Bearing Tester can not accept most of the needed instrumentation for the support or thrust bearings. The single piece housing design is too restricted by existing ports and retainer bolts to allow new access for instrumentation. Furthermore, existing instrumentation should be moved prior to initiating any test program.

3.2.4 Test Objectives

The OTV Bearing Tester has an Invar housing and titanium shaft. This combination would require calculation of clearance profiles which may result in suspect experimental data.

3.2.4.1 Flow Field Measurement. Detailed mapping of the bearing pressure and temperature profiles in the bore is not possible. There is no access for instrumentation leads or transducers. This limitation would severely limit the ability to perform this test for any of the bearing configurations.

Overall, this tester is not capable of any flow field pressure or temperature mapping.

3.2.4.2 Bearing Performance Measurement. The measurement of leakage is very straight forward for externally fed hydrostatic bearings and other bearings fed through internal housing annuli. Difficulties arise for other bearing configurations which would leak into the radial loader and thrust bearing cavities. Leakage measurements for these configurations will always have uncertainty due to the mixing of the flows and requirement to subtract inlet quantities from drain measurements.

The determination of load capacity is difficult. Since the loader is fixed in the housing, its load value can only be determined by calculation using measured quantities for displacement, pressure, and temperature. Therefore load capacity data is not valid for validation of analytical predictions.

Accurate measurement of torque for the support bearings is impossible. Too many load paths exist by way of the thrust bearings and radial loader for an accurate assessment of individual bearing torque to be made. There is also large uncertainty in the power supplied by the turbine.

Overall, this tester is not capable of meeting the requirements for bearing performance measurement. Its poor and inflexible design prevent useful measurement of basic quantities.

3.2.4.3 Simulation Of Rotordynamic Characteristics. The OTV Bearing Tester can perform rotordynamic simulations for any turbomachine with a cylindrical shaft with uniform mass and inertia distribution. Which is another way to say that it is not useful at all. No space is available at either end to add extra mass or inertia to simulate impellers.

3.2.4.4 Extraction Of Rotordynamic Coefficients. Techniques utilizing change in unbalance of the shaft can be used to measure effective stiffness and effective damping with this tester. Separated rotordynamic coefficients can not be obtained because force measurements can not be made and shaft excitation is limited to synchronous frequencies.

3.2.4.5 Transient Start/Shutdown Simulation. This tester was supposedly designed for transient start/shutdown simulation. However, the fixed radial load device effectively prevents control of radial load as a function of shaft speed. In order for this tester to be useful, a new housing would be required which would accept a free moving radial load device.

3.2.4.6 Test Objective Summary. The OTV Bearing Tester is not capable of performing any of the required testing functions without a virtual redesign of the entire tester.

3.2.5 OTV Bearing Tester Summary

Based on the aforementioned discussion, it can be concluded that the OTV Bearing Tester is far too inflexible to be considered useful as a research tool for fluid film bearings. It can not accommodate the proper instrumentation, its basic thrust control and radial load functions are suspect, it is not LOX compatible, and its housing can not be modified to improve its performance.

3.3 LONG LIFE BEARING TESTER

The Long Life Bearing Tester was originally designed for liquid hydrogen testing and has the following capabilities:

- 500 lbf. radial load capability
- 3000 psi supply pressure

- 800 psi sump pressure
- 50,000 rpm turbine drive

3.3.1 Test Article Configurations

This tester has one test bearing location. The slave bearing is a duplex ball bearing which can not be easily replaced with a fluid film bearing as the slave bearing reacts all of the thrust load.

3.3.1.1 Hydrostatic Bearings. The tester was originally designed for both externally and internally fed hydrostatic bearings. Therefore, the basic construction is well suited to their testing. Practical hydrostatic bearing diameters are in the range 50-90 mm. Bearing length is limited to approximately 50 mm. Therefore, bearing L/D could be varied between 0.0 and 1.0, which is acceptable but limiting for most hydrostatic bearing applications.

For internally fed hydrostatic bearings, the tester has a separate shaft with slots. High pressure fluid is supplied in the cavity between the slave and test bearing. The pressure is contained using two sets of carbon ring seals. The fluid is supplied to the test bearing by way of two rows of holes in the main shaft which lead to an annulus between the shaft and journal sleeve (as shown in Figure 3-3). From the annulus, the fluid travels into the hydrostatic bearing orifice. This tortuous path will result in a significant loss of supply pressure. For future testing, the flow path design should be optimized for minimum supply pressure loss.

3.3.1.2 Annular Seals. The LLBT can and has been configured to test annular seals. The axial thrust can be balanced, however, no measurements of inlet velocity or leakage can be made. Therefore, it can not be used for the validation of analytical models and is only useful for demonstration purposes or material wear studies.

3.3.1.3 Foil Bearings. Both types of foil bearings (bending and tension) can be inserted into the tester. However, the L/D range of 0.0-1.0 is somewhat restrictive for the bending foil bearing. Adequate radial space is available for the tension foil bearing.

3.3.1.4 Hydrodynamic Bearings. Fixed geometry and tilt pad hydrodynamic bearings of all types should easily fit into this tester. The radial load capability is more than enough to fully exercise all bearing configurations.

3.3.1.5 Hybrid Fluid Film Bearings. The available bearing envelope of 50-90 mm diameter and 50 mm length should be sufficient to accommodate all of the currently proposed hybrid fluid film bearing concepts.

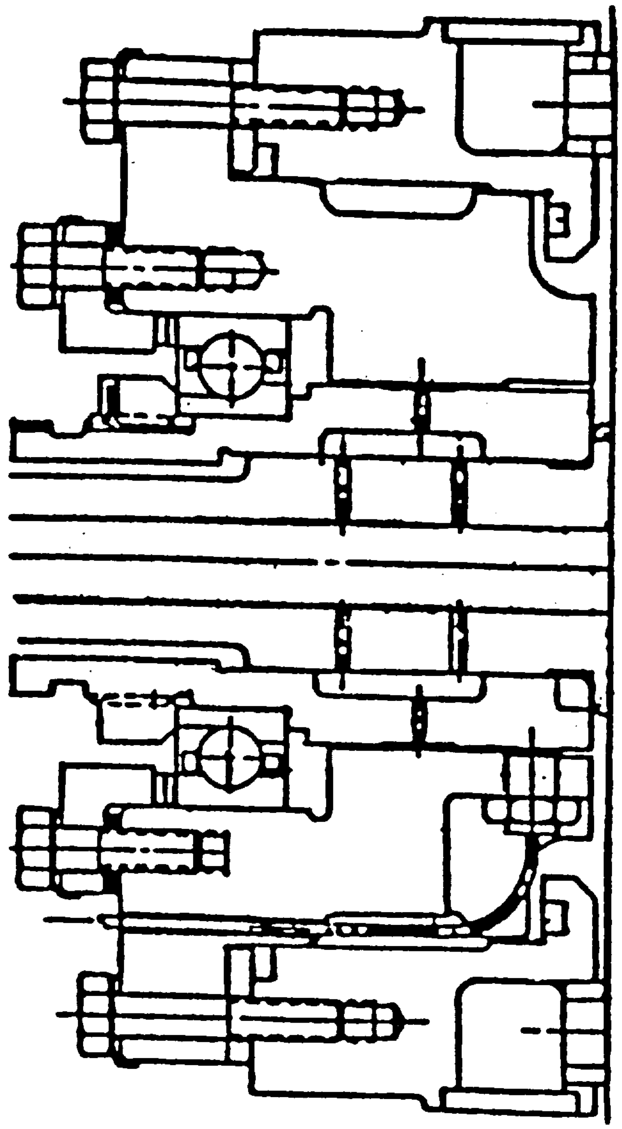


Figure 3-3. Long Life Bearing Tester with and internally fed hydrostatic bearing

3.3.1.6 Hybrid Magnetic Bearings. Referring to the information provided in Figure 2-8, it appears that it would be straightforward to accommodate magnetic bearings and their hybrids in a reasonable size and load capacity. The high speed of the tester would require a laminated shaft which could be easily accomplished using a shaft sleeve.

3.3.1.7 Thrust Bearings. This tester currently reacts all axial thrust with a duplex pair of ball bearings. Replacement of this bearing with an alternate thrust bearing is not feasible. A thrust bearing could be configured to operate on the test bearing end of the shaft. This would also require the development of an axial load device to load the bearing.

3.3.1.8 Test Article Summary. The discussions presented above reveal that within the envelope afforded by the test bearing location that all conceivable conventional and hybrid fluid film bearings can be accommodated. Thrust bearing testing can be accomplished by developing an axial load device and hanging a runner off of the free end of the shaft. Annular seal testing is not feasible due to the inability to measure quantities fundamental to its performance.

3.3.2 Test Fluids

Since the tester was designed for liquid hydrogen and liquid oxygen use, has been tested in all desired fluids, and currently meets all pressure requirements, there are no issues related to this topic.

3.3.3 Instrumentation

The ability of this tester to accept all potential test articles is not matched when it comes to instrumentation. The test article carrier is also the high pressure fluid supply tube. This prevents radial access to the test article because the supply annulus can not be punctured. Some instrumentation can be taken out axially, but is very limited.

3.3.3.1 Pressure Measurement. Supply pressure measurements for this tester in the externally pressurized configuration are made at the fluid inlet to the housing which is somewhat remote from the test article. The fluid pressure is altered between this point and the test article by bends and annuli. For internally pressurized bearings, the fluid travels a tortuous path resulting in an unquantifiable supply pressure. The accuracy of this measurement would be improved using a slip ring assembly and redesigning the supply system. Pressure measurements in the bearing clearance can be made in limited numbers.

3.3.3.2 Temperature Measurement. Like pressure measurements, temperature measurements can be made in limited quantity and are of questionable accuracy due to the somewhat remote location of the measurement. This uncertainty would make it difficult to obtain repeatable conditions for two-phase flow tests.

3.3.3.3 Acceleration Measurement. Plenty of surface area is available for housing acceleration measurements, both radially and axially. Internal to the tester accelerometer mounting is readily available on the test bearing carrier.

3.3.3.4 Force Measurement. Force measurements can be made on the bearing carrier, however, extensive calibrations are required to eliminate the influence of the fluid supply tube stiffness.

3.3.3.5 Torque Measurement. Torque measurement for the test bearing is not possible with conventional instrumentation. Exotic methods such as strain measurements using an embedded magnetic strip would be required to obtain even crude estimates.

3.3.3.6 Proximity Probes. Access for a single plane of proximity probes is already available. However, it would be very difficult to install a second plane of proximity probes on the back side of the test article without major modifications and severe limitations on available test bearing envelope. Unfortunately, this is a major disadvantage since this tester is prone to excessive tilt at the test article which must be measured for validation of analytical models.

3.3.3.7 Slip Rings And Transmitters. A slip ring assembly could be configured to feed out of the free end of the shaft. Shaft bore size is too limited to accept transmitters.

3.3.3.8 Instrumentation Summary. The Long Life Bearing Tester has questionable instrumentation in its current configuration and is severely limited in its ability to accept more instrumentation by design constraints on the test bearing carrier. The tester design can be modified to accept a slip ring at the free end of the shaft enabling measurements internal to the shaft.

3.3.4 Test Objectives

3.3.4.1 Flow Field Measurement. Detailed mapping of the bearing pressure and temperature profiles in the bore is not very likely due to the lack of access for a significant number of pressure and temperature measurements. No velocity measurements in the bore are possible due to lack of radial access across the fluid supply annulus. This limitation severely limits the ability to test annular seals. The lack of radial access also eliminates the possibility of performing flow visualization tests.

Overall, this tester is not capable of extensive flow field pressure and temperature mapping for the bearing configurations of interest.

3.3.4.2 Bearing Performance Measurement. The measurement of leakage is very straight forward for externally fed hydrostatic bearings and other bearings fed through internal housing annuli. Difficulties arise for internally fed hydrostatic bearings and other bearing configurations which rely on the ring seals for containment. Leakage measurements for these configurations will always have uncertainties of the same magnitude as the seal leakage. With so many common drains, the seal leakage can not be measured accurately, it must be calculated based on pressure measurements.

The determination of load capacity is fairly straightforward. The radial loader is currently instrumented and calibrated with strain gauges and can easily be fitted with force transducers to improve load accuracy. The bearing carrier can easily be mounted in force transducers for direct bearing reaction load measurements.

Accurate measurement of torque for the test bearing is virtually impossible. The fluid supply tubes are much to stiff for a measurement on the bearing carrier. As mentioned before, exotic methods could be employed but there is little chance of success.

Overall, this tester is relatively capable meeting the requirements for bearing performance measurement. It has advantages in load capacity measurement but falters for leakage measurement in some configurations. It is not capable of torque measurement. It should be noted that all measurements made with this tester are suspect unless accurate measurement is made of shaft tilt.

3.3.4.3 Simulation Of Rotordynamic Characteristics. The Long Life Bearing Tester is not very well suited to performing rotordynamic simulations due to the presence of a duplex pair of rolling element bearings with deadband at the slave bearing location. This results in a highly nonlinear system for which reliable analytical predictions are difficult to obtain. The dynamic characteristics of the test article have a significant influence on the dynamic performance of the slave bearings which prevents a direct comparison of different bearing geometries and configurations under constant operating conditions.

3.3.4.4 Extraction Of Rotordynamic Coefficients. In its original configuration the LLBT was not capable of obtaining data for the determination of rotordynamic coefficients. Alternative configurations of the tester have been developed for other NASA contracts which eliminate many of the shortcomings related to the original design. However, all practical methods for exciting the shaft impart tilt to the shaft which must be accurately measured. Additionally, the tester must undergo extensive calibration in order to eliminate the influence of the fluid supply tube stiffness on the dynamic force measurement. Therefore, it is unlikely that useful rotordynamic coefficient measurements can be made with this tester.

3.3.4.5 Transient Start/Shutdown Simulation. This tester has already been used somewhat successfully for transient start/shutdown simulation in liquid nitrogen and liquid oxygen. Radial loader capacity is sufficient to simulate virtually any load quantity and profile which is characteristic of turbopump operation. Again, the issue with utilizing this tester is the control and measurement of tilt. For transient testing it is more important to eliminate tilt than to measure it in order to insure shaft-housing material contact which accurately simulates the turbopump condition. It is unlikely that this tester can be redesigned to eliminate shaft tilt.

3.3.4.6 Test Objective Summary. The Long Life Bearing Tester is not very capable of performing the required testing functions due to limitations in instrumentation access and fundamental problems with shaft tilt.

3.3.5 Long Life Bearing Tester Summary

Based on the aforementioned discussion, it can be concluded that the Long Life Bearing Tester can test all bearing configurations, but can test none of them well. The tester is compatible will all desired fluids and has already demonstrated compatibility in test. However, existing instrumentation is of suspect value and access for more and improved instrumentation is limited by the basic design of the tester. The lack of proper instrumentation limits the ability of the tester to meet required test objectives. Although the tester can easily accommodate extensive dynamic instrumentation, it is not suitable for dynamic testing due to the presence of rolling element slave bearings with deadband, supply tube stiffness, and shaft tilt.

3.4 TECHNICAL EVALUATION SUMMARY

Based on the aforementioned discussions, the following conclusions can be drawn about the suitability of each tester to the main objective of validation of analytical models:

- The Hybrid Bearing Tester is very versatile with respect to accommodating most, but not all, bearing configurations, extensive amounts of instrumentation, and satisfying test objectives. Design improvements can be made by changing housing material (LOX compatibility), improving maximum pressure capability, and creating new shaft/housing parts for internal flow capability.

- The OTV Bearing Tester is better used as a boat anchor than a fluid film bearing tester. It is very limited with respect to bearing configuration installation, instrumentation, and meeting test objectives. Improvement in the design would require a completely new tester cross-section.

- The Long Life Bearing Tester can accommodate all foreseeable bearing configurations. However, instrumentation access is limited and the design suffers from shaft tilt. Design improvements to eliminate the shaft tilt are difficult because the shaft naturally pivots from the slave bearings and the loader is located at the test bearing instead of being centrally located.

4.0 DESIGN IMPROVEMENTS

4.1 HYBRID BEARING TESTER

As mentioned previously, the major design issues facing the HBT are:

- LOX compatibility
- Low supply pressure
- Clearance control at cryogenic temperatures
- Elimination of ring seals in loader cavity
- Increased radial loader capability

Other design improvements which would expand the capability of the tester are:

- Use of radial loader cavity for test bearing
- Access for additional radial loader
- Hollow shaft for internally fed bearings
- Shaft with added mass and inertia for rotordynamic simulation

4.1.1 Major Design Improvements

The major design issues can be addressed with two efforts: redesign the housing to be manufactured out of Inconel 718 and change the radial loader to a rolling element design which would not require ring seals.

The first task would solve the LOX compatibility and clearance control issues. Inconel 718 has proven LOX compatibility and making the housing the same material as the shaft will alleviate most thermal expansion issues. The low supply pressure problem would also be addressed because the Inconel 718 has a yield strength which is about 50% higher at cryogenic temperatures than that of 316 SS. Revamping the fluid supply system in the bearing carriers to eliminate the weld points and completing the proper stress analysis would result in a much more robust design.

The second task is the one which will significantly expand the capability of the tester. The assembly procedure requires that the bearing on the free end of the tester have a diameter less than that of the ring seals so that the bearing carrier (which has the ring seal assembly bolted to it) can slide over the already installed shaft. Without the ring seals the L/D range can be expanded from 0-0.67 to 0-1.5. This enables testing of realistic bearing geometries for hydrostatic, hydrodynamic, and foil bearings. Changing the loader to a rolling element bearing design (shown in Figure 4-1) eliminates the ring seals, provides consistent loading, and increases the loader capability. This type of loader will also function better with dynamic shakers or impactors than a hydrostatic load shoe. The sealing for the new loader configuration can be accomplished by providing a single ring (or brush) seal at the pass through.

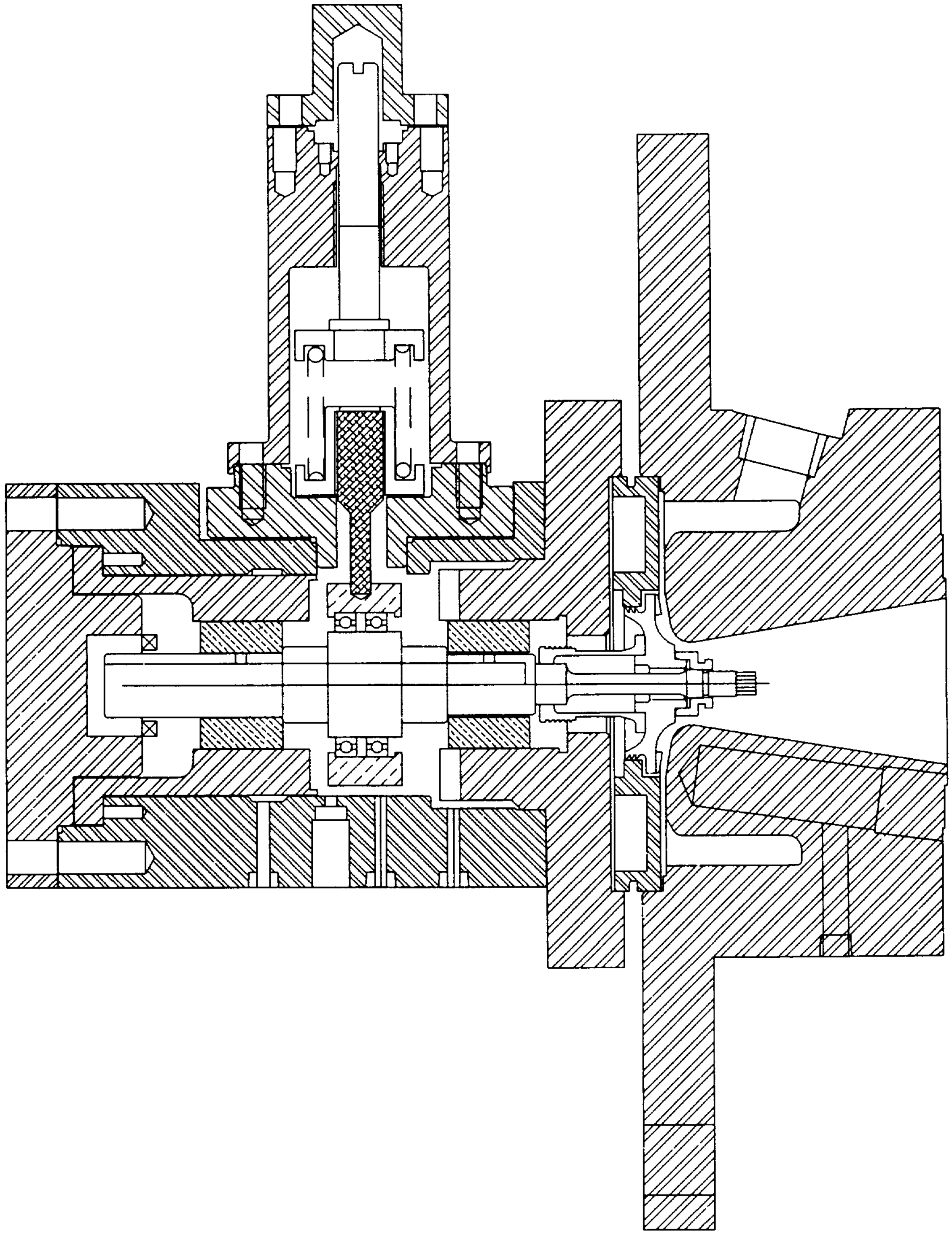


Figure 4-1. Hybrid Bearing Tester configured with rolling element bearing loader

4.1.2 Other Design Improvements

The other design improvements listed above are not critical items. Instead, they are tasks which either further increase the type and range of bearings which can be tested or enable additional test objectives to be met.

Use of the radial loader cavity for test bearing installation promises to significantly expand the range of bearing sizes which can be accommodated. Using this space will allow the testing of large hydrodynamic bearings such as tilt pad bearings, magnetic bearings, and some hybrid configurations. It will also significantly expand tester capability by increasing access for test article instrumentation, enabling the measurement of torque, and enabling the extraction of rotordynamic coefficients. The addition of a second radial loader also addresses the issue of rotordynamic coefficient extraction as well as the testing of asymmetric bearing geometries without disassembly and retesting to obtain different load angles.

The addition of a hollow shaft would enable the testing of internally fed hydrostatic bearings. With the addition of a slip ring assembly, the hollow shaft could also be used to map internal pressure and temperature profiles for bearings which have difficult to access fluid films such as foil bearings.

Finally, a new shaft assembly could be designed with additional features to simulate the mass and inertia properties of impellers for rotordynamic simulation testing.

4.2 OTV Bearing Tester

The OTV Bearing Tester is so limited in its capability that only a complete redesign of the basic features of the tester housing with a new LOX compatible material combined with a new LOX compatible shaft material will result in the type of tester needed for this effort. In other words, it is not worth reviewing since a new tester could be designed from scratch in less time and for less cost than it would take to salvage the OTV Bearing Tester.

4.3 Long Life Bearing Tester

As mentioned previously, the major design issues facing the LLBT are:

- Shaft tilt
- Fluid supply tube/annulus
- Rolling element slave bearing

Other design improvements which would improve performance of the tester are:

- Improved flow path for internally fed bearings
- Eliminate fluid supply tube stiffness

4.3.1 Major Design Improvements

The major design issues of shaft tilt and the rolling element slave bearing are difficult to address. The limitations related to the fluid supply tube/annulus can be addressed by designing a new bearing carrier which will be press fit into the housing.

Although the LLBT has many configurations for support bearing location and radial load application, the problem of shaft tilt remains with each. The original configuration of the tester had rolling element slave bearings located just inside of the test bearing. This configuration leaves the test bearing overhung. Radial loads were applied on the outside of the carrier or to the supply tube support. Even with the shaft fixed, the potential for shaft/housing relative tilt is high because the bearing carrier is cantilevered. Application of a radial load to the carrier would deflect the supply tubes and not result in pure translation. The tester was later revised to perform start transient testing. This revision involved extending the shaft and adding a rolling element to the radial load mechanism so that load could be applied directly to the shaft. In this configuration the test bearing also functions as a shaft support. Load applied to the shaft causes tilt due to the pivoting action of the shaft in the slave bearing. It is not readily apparent how to modify this tester to eliminate shaft tilt in either configuration.

The limitation imposed by the fluid supply tube/annulus could be removed by redesigning the bearing carrier to be more like the ones in the HBT. The current bearing carrier is supported by tubes which are connected to a housing flange. This flange could be discarded and a new flange designed with a more rigid structure. This new cylindrical bearing carrier would be press fit into the housing. Supply ports for hydrostatic bearing testing could be routed through the new support in a way which would leave plenty of room for pressure and temperature measurements. Another alternative is to have a free floating bearing carrier which is supported solely by load cells. This would enable the measurement of torque, improve rotordynamic coefficient extraction, and allow access for pressure and temperature measurements. Both concepts are shown schematically in Figure 4-2.

The rolling element slave bearing is not easily replaced or improved. This bearing reacts all of the thrust load. Furthermore, the high tester speed requires that deadband be present to accommodate thermal growth of the races and prevent bearing failure.

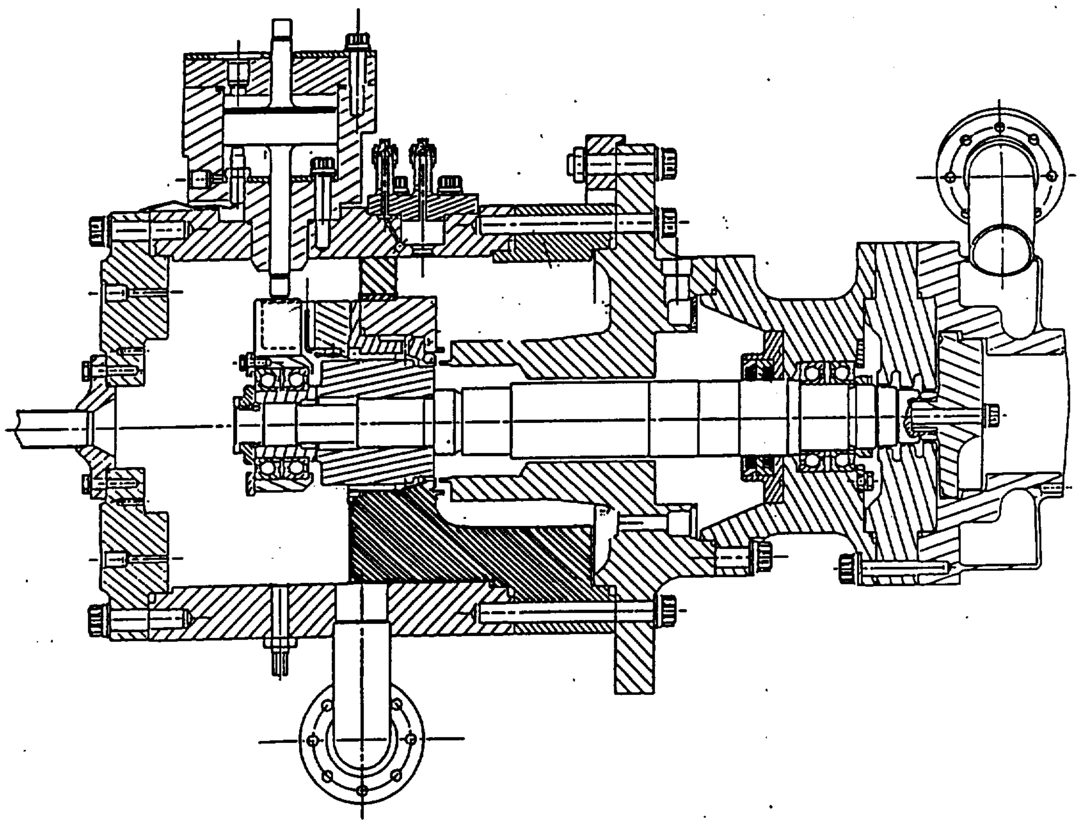


Figure 4-2. Long Life Bearing Tester bearing support concepts

4.3.2 Other Design Improvements

The other design improvements listed above are not critical items. Instead, they are tasks which either further increase the type and range of bearings which can be tested or enable additional test objectives to be met.

Use of the radial loader cavity for test bearing installation promises to significantly expand the range of bearing sizes which can be accommodated. Using this space will allow the testing of large hydrodynamic bearings such as tilt pad bearings, magnetic bearings, and some hybrid configurations. It will also significantly expand tester capability by increasing access for test article instrumentation, enabling the measurement of torque, and enabling the extraction of rotordynamic coefficients. The addition of a second radial loader also addresses the issue of rotordynamic coefficient extraction as well as the testing of asymmetric bearing geometries without disassembly and retesting to obtain different load angles.

The addition of a hollow shaft would enable the testing of internally fed hydrostatic bearings. With the addition of a slip ring assembly, the hollow shaft could also be used to map internal pressure and temperature profiles for bearings which have difficult to access fluid films such as foil bearings.

Finally, a new shaft assembly could be designed with additional features to simulate the mass and inertia properties of impellers for rotordynamic simulation testing.

5.0 COST AND SCHEDULE FACTORS

The preceding discussions have centered on the technical capabilities of the candidate testers without regard to cost and development time. While the technical evaluation is necessary for determining the suitability of a particular design for the application, cost and schedule considerations are typically the primary factors used to discriminate between design candidates and to determine feasibility of the project.

5.1 HYBRID BEARING TESTER

As discussed above, the two design improvements needed to enable this tester to meet the stated criteria are a new housing and a new radial loader.

5.1.1 Cost

The estimated costs to complete these two design tasks and fabricate the components are shown in Table 5-1.

Table 5-1. HBT Design Costs

	HOUSING		LOADER	
TASK	Manhours	Dollars	Manhours	Dollars
Conceptual Design	40		20	
Detailed Design	160		120	
Stress Analysis	160		120	
Hydrodynamic Analysis	40		40	
Bearing Analysis	10		60	
Fabrication		$35,000		$10,000
TOTAL	410	$35,000	360	$10,000

5.1.1.1 <u>Housing Design Costs.</u> Other than change the material for LOX compatibility and clearance control, the goals of the housing design change include increasing the supply pressure capability of the tester. An increase in pressure capability from 650 psi to 2000 psi will require careful analysis of the stresses in the parts. Also, the increased flowrate may dictate an increase in drain port diameter. Calculation of leakage for typical bearing configurations will need to be made in order to assess flow requirements.

5.1.1.2 Loader Design Costs. The loader design change will significantly increase the loads on the housing and shaft through increased load capacity and cavity pressure. Elimination of the ring seals will significantly increase the pressure in the loader cavity resulting in larger stresses on the loader housing flange bolts. The large steady and dynamic loads will require careful design and selection of the rolling element bearings.

5.1.2 Schedule

The estimated schedule for the design improvements is shown in Figure 5-1. The figure shows that the housing design effort is estimated to take 3 months to complete and the loader design 2.5 months. Fabrication times for the two efforts are the same as the design time.

5.2 OTV BEARING TESTER

As discussed in the technical review, substantial improvement in tester capability through modification of the existing design is not likely. Therefore, no estimates are provided. A completely new tester would result in a more useful research tool and cost approximately 4000 manhours to design, $100,000 to fabricate, and take 1 year to complete.

5.3 LONG LIFE BEARING TESTER

As discussed above, the only design improvement which will substantially improve the versatility and utility of the LLBT is a redesign of the bearing carrier to be free floating. This will eliminate the supply tube/annulus restrictions and shaft tilt issues.

5.3.1 Cost

The estimated costs to complete these this design task and fabricate the components are shown in Table 5-2.

5.3.1.1 Redesign Costs. Redesigning the bearing carrier to be free floating is not a trivial task. This change impacts the radial loader, shaft, and housing designs. The main housing will have to be redesigned to allow access for various support/measurement options which will depend on the test objective. The radial loader will need to be reconfigured to function as load, anti-rotating, and possible steady-state support. The shaft will have to be redesigned to eliminate the current loader assembly and be stiffened to withstand the high concentrated loads which will result from installing the rolling element slave bearings. The extensive nature of this redesign and the inclusion of shaft/bearing changes will require rotordynamic analysis to verify speed ranges for safe operation.

Figure 5-1. Hybrid Bearing Tester design schedule

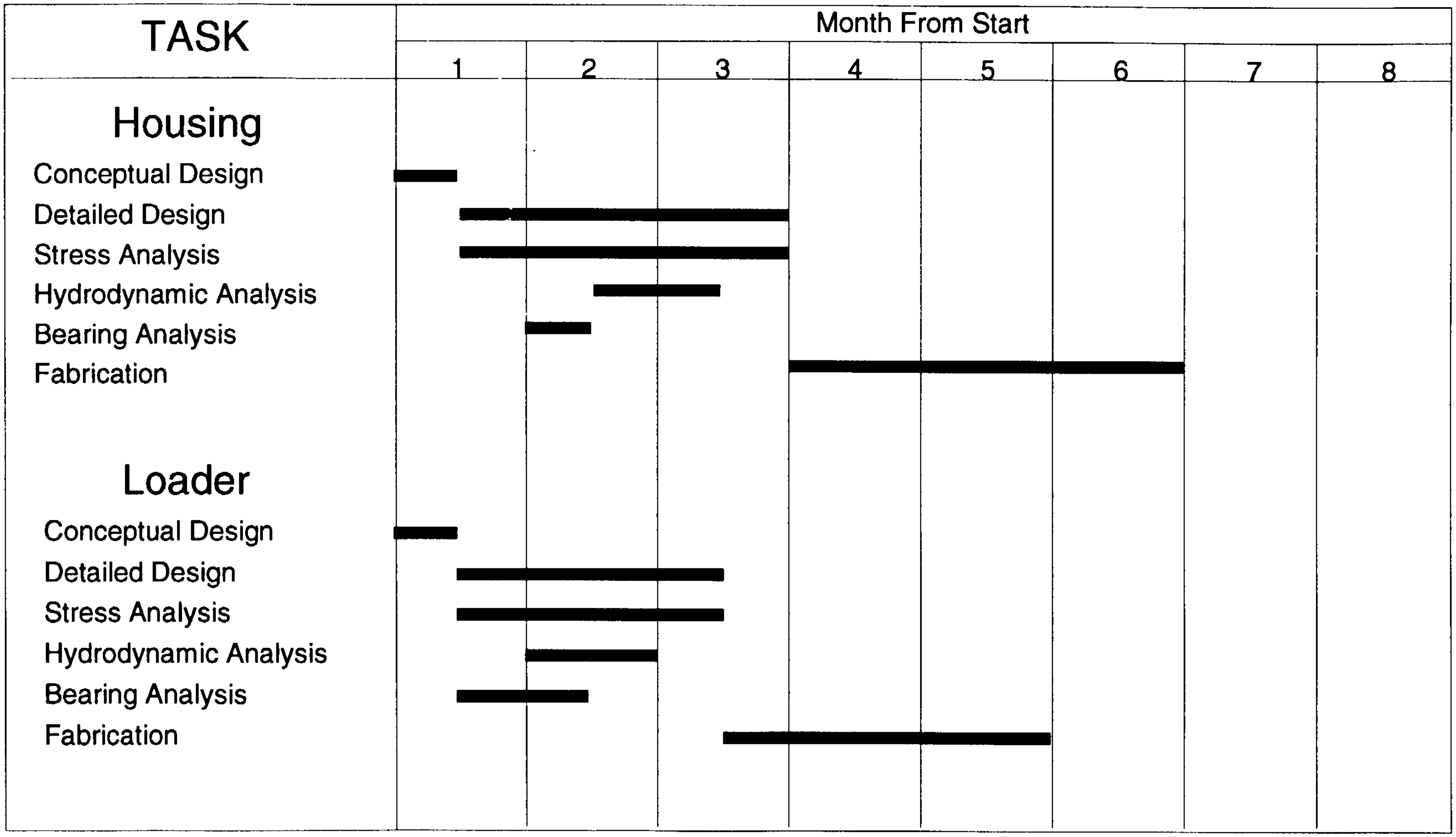


Table 5-2. LLBT Design Costs

TASK	Manhours	Dollars
Conceptual Design	80	
Detailed Design	300	
Stress Analysis	200	
Hydrodynamic Analysis	80	
Rotordynamic Analysis	80	
Bearing Analysis	60	
Housing Fabrication		$25,000
Loader(s) Fabrication		$10,000
Shaft Fabrication		$15,000
TOTAL	800	$50,000

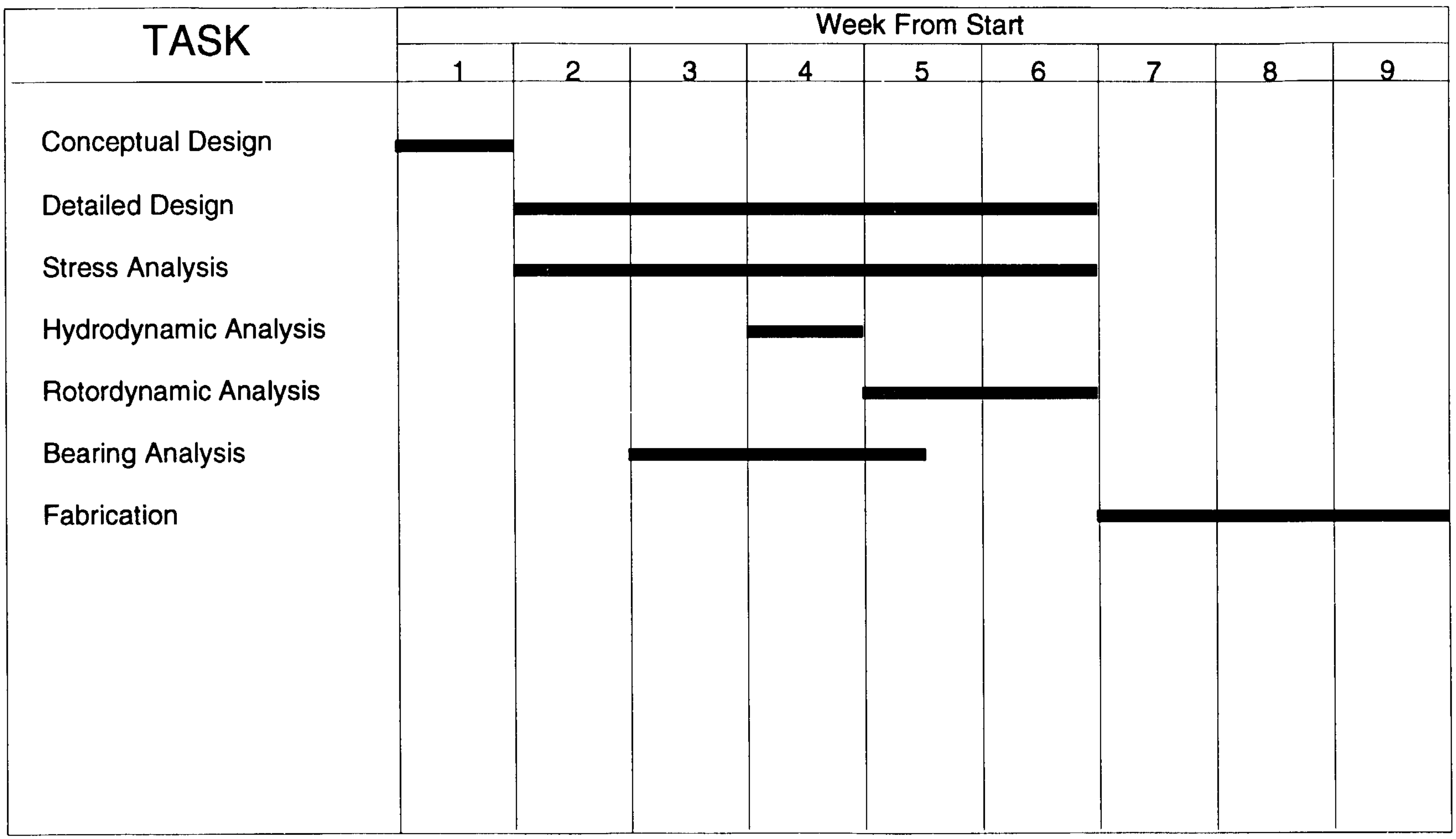
5.3.2 Schedule

The estimated schedule for the design improvements is shown in Figure 5-2. The figure shows that the design effort is estimated to take 6 months to complete. Fabrication times for the effort is estimated at 3 months.

5.4 SUMMARY

The data presented above shows that the same amount of effort is required to bring both testers up to maximum capability. The main difference is that the HBT can be changed incrementally while the LLBT must undergo extensive redesign of every critical feature (housing, shaft, and loader) simultaneously. From a project risk perspective it is better to proceed with the HBT. The radical changes required for the LLBT will result in increased shakedown testing costs and possible redesign.

Figure 5-2. Long Life Bearing Tester design schedule



6.0 RECOMMENDATIONS

Based on a review of all of the technical, cost, and schedule factors presented, it is recommended that the Hybrid Bearing Tester be selected for use a general purpose fluid film bearing tester. The basis for this recommendation is as follows:

- Although it falls short of the LLBT in its ability to accept all possible bearing configurations, the HBT is far superior in the one catagory that matters the most: the ability to provide accurate data for validation of analytical models.
- The HBT can be used immediately for some test objectives with a limited number of bearing configurations while the LLBT has serious problems with shaft tilt and restricted access that limit its usefulness.
- The HBT has thrust bearing test capabilities whereas the LLBT has none.
- The changes needed to improve the HBT are less risky than those required for the LLBT.
- The funding required to improve the HBT can be provided incrementally.

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16. Abstract

The designs of three existing testers (Hybrid Bearing Tester, OTV Bearing Tester, and Long Life Bearing Tester) owned by NASA were reviewed for their capability to serve as a multi-purpose cryogenic fluid film bearing tester. The primary tester function is the validation of analytical predictions for fluid film bearing steady-state and dynamic performance. Evaluation criteria were established for test bearing configurations, test fluids, instrumentation, and test objectives. Each tester was evaluated with respect to these criteria. A determination was made of design improvements which would allow the testers to meet the stated criteria. The cost and time required to make the design changes were estimated. A recommendation based on the results of this study was made to proceed with the Hybrid Bearing Tester.

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